EXCLUSIVE technical and non-technical articles on design, construction and operation of oil-engines and motorships by the world's foremost writers on marine engineering.

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Ohio River Pipeline Dredge Now in Service

New Vessel for the United States Engineer Office, Cincinnati, Has Direct Connected Diesel Dredge Pump

BY GEO. SZEPINSKI

WITH the commissioning of the United States dredge C. B. HARRIS by the United States Engineering Office, Cincinnati, the first of the modern pipeline dredges which the War Department is building has gone into service on an inland waterway. This new 24-in. pipeline dredge embodies the first installation of a direct-connected Diesel dredge pump and forms an interesting contrast with the Diesel electric type. The Charles Ward Engineering Works of Charleston, W. Va., were responsible for this vessel, having built the hull, installed the machinery and fitted out the dredge.

The hull is 175 ft. long, 50 ft. wide and 8 ft. deep, with a draft of 5 ft. 2½ in.; a steel deck house 10 ft. high, 40 ft. wide and 132 ft. long, houses the pumping machinery and all auxiliaries.

Above the upper deck are the living quarters of the officers and the crew in a steel deck house of approximately the same dimensions as the house below this deck, but with a deck height of 8 ft. Both decks, main and upper, are straight, without camber or sheer, but the roof over the quarters is built with the conventional camber to keep it properly drained.

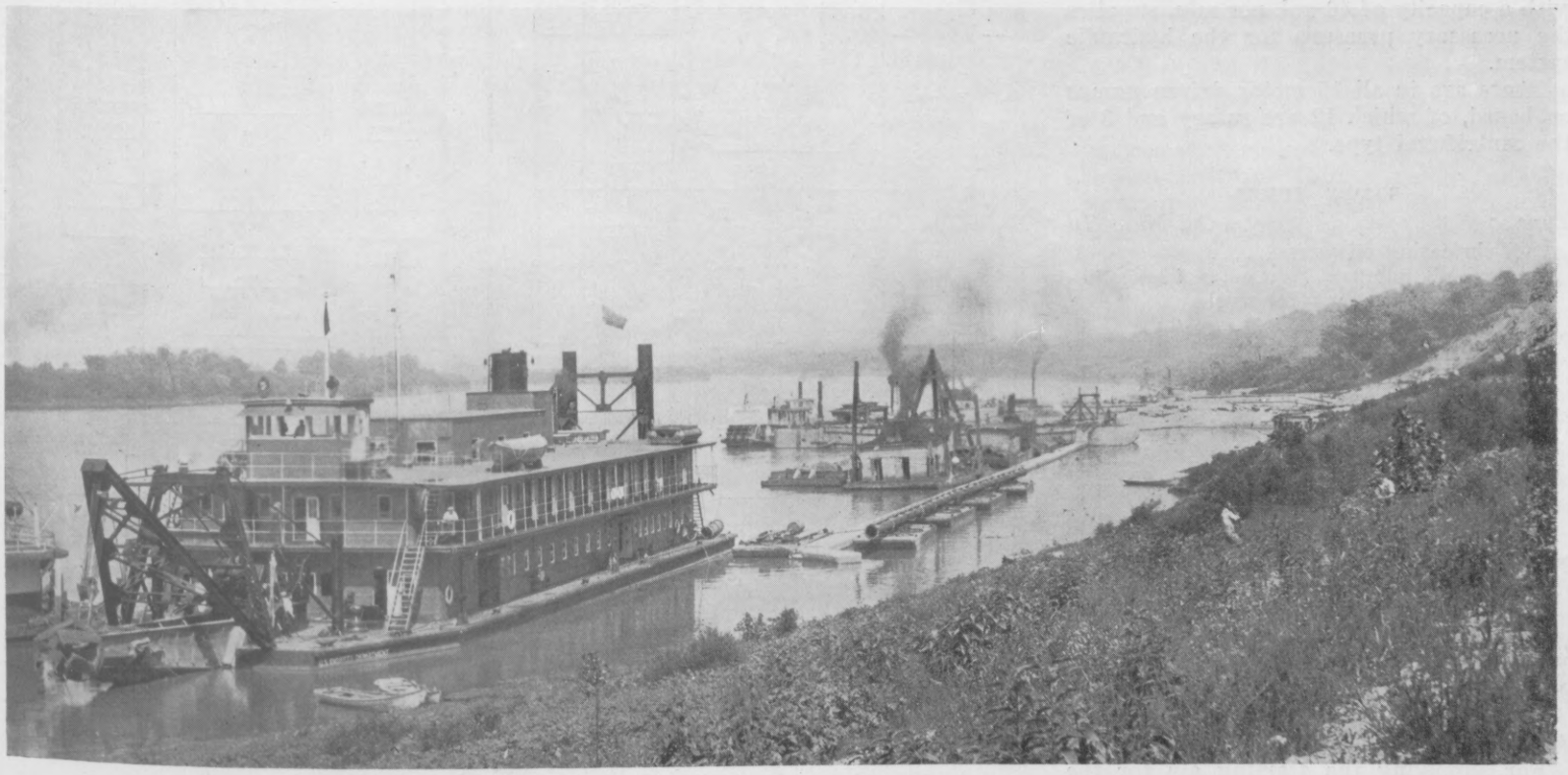
The superstructure above the top of the deck house contains the lever house at the ladder end of the dredge and the engine casing with muffler house at the spud end. A dummy stack, housing the three exhaust pipes, is built on top of the muffler house.

The 1000 b.hp. main Diesel engine, built by the McIntosh & Seymour Corporation, is of the six-cylinder type and drives the 8-ft pump impeller through an 11¾-in. diameter shaft, screwed into the impeller and coupled to the engine through a flexible coupling. The impeller shaft is supported by two self-aligning roller-type bearings and the thrust of the impeller is taken up by a deep-groove ball thrust bearing. As the pump is located above the water line a water primer is installed on top of the pump housing of sufficient size to prime the pump in less than two and one-half minutes.

The two auxiliary engines, which are also of the McIntosh & Seymour Corporation's make, develop 225 b. hp. and are each direct-connected to a 150 kw. generator, one of which is sufficient to furnish the current for the auxiliaries described further on and for the hoisting and hauling winch motor, which is a 50 hp. compound-wound motor fitted with a reversible type controller providing for 5 speeds in each direction. A bakelite pinion is fitted to the motor shaft, and by means of a lay shaft and clutches power may be applied to the five drums of the winch, two for the spuds, two for hauling port and starboard and one for hoisting and lowering of the dredging ladder. A winch head is placed on each shaft end, projecting through the sides of the house and intended for use in breasting.

The two outboard drums are arranged for the hoisting and lowering of the spuds, the rope from the drum leading through the hull over fairleads to the spud frame at the after end of the dredge. Each spud weighs about 11 short tons, is of circular cross-section, 56 ft. 9 in. long and built up of plates and angles. Two gates, about 11 ft. 10 in. apart guide each spud.

The hauling drums, directly inboard of



Diesel engined dredge, C. B. HARRIS, ready to hook up to a long pipe at Rising Sun, Ind.

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the spud drums, each carry 1000 ft. of one-inch wire rope for swinging the dredge during operation, and the center drum operates the one-inch rope tackle used for hoisting and lowering the ladder, there being five extra-heavy 30-in. metallic bushed sheaves on the A-frame at the bow of the dredge and four similar sheaves on the ladder end.

A very substantial frame, built up of plates and angles, supports the hinged suction-pipe that leads to the cutter-head. Near the pivot of the ladder the cutter motor is fastened. This is a totally inclosed watertight motor with compound winding arranged at such proportions that the speed will drop from 400 r.p.m. at 150 hp. to 325 r.p.m. at 200 hp. The no-load speed of the motor is 600 r.p.m. One self-aligning radial roller bearing at the pinion end and one deep groove radial ball bearing at the commutator end support the armature and allow for operating the cutter with armature shaft inclined as much as 45 deg. A contactor type controller with time limit acceleration and stalling feature is provided; also the necessary switches, jamming and timing relays.

In the lever house, we find the master controller for the cutter motor, also the control of the five clutches and five brakes for the hoisting and hauling winch motor. All controls are arranged in a compact and convenient manner for operation. Two stands, one on each side of the operator, are provided, on which are mounted the 10 handles for the clutches and brakes; in front of the operating stand are the motor controls. The brake and clutch handles are attached to extension stems which reach below the upper deck and operate the cocks regulating the water supply for the hydraulic system. This system consists of 10 cylinders fitted in place over the clutches and brakes, which are worked by applying the pressure on the end of the lever for the clutches and by lifting and lowering a weight acting on the brake band lever. A motor driven rotary pump with a capacity of 40 gal. per min. supplies the necessary pressure for the hydraulic system.

There are in all 15 motor driven pumps on board, of which 12 are rotary and 3 of the centrifugal type.

ROTARY PUMPS

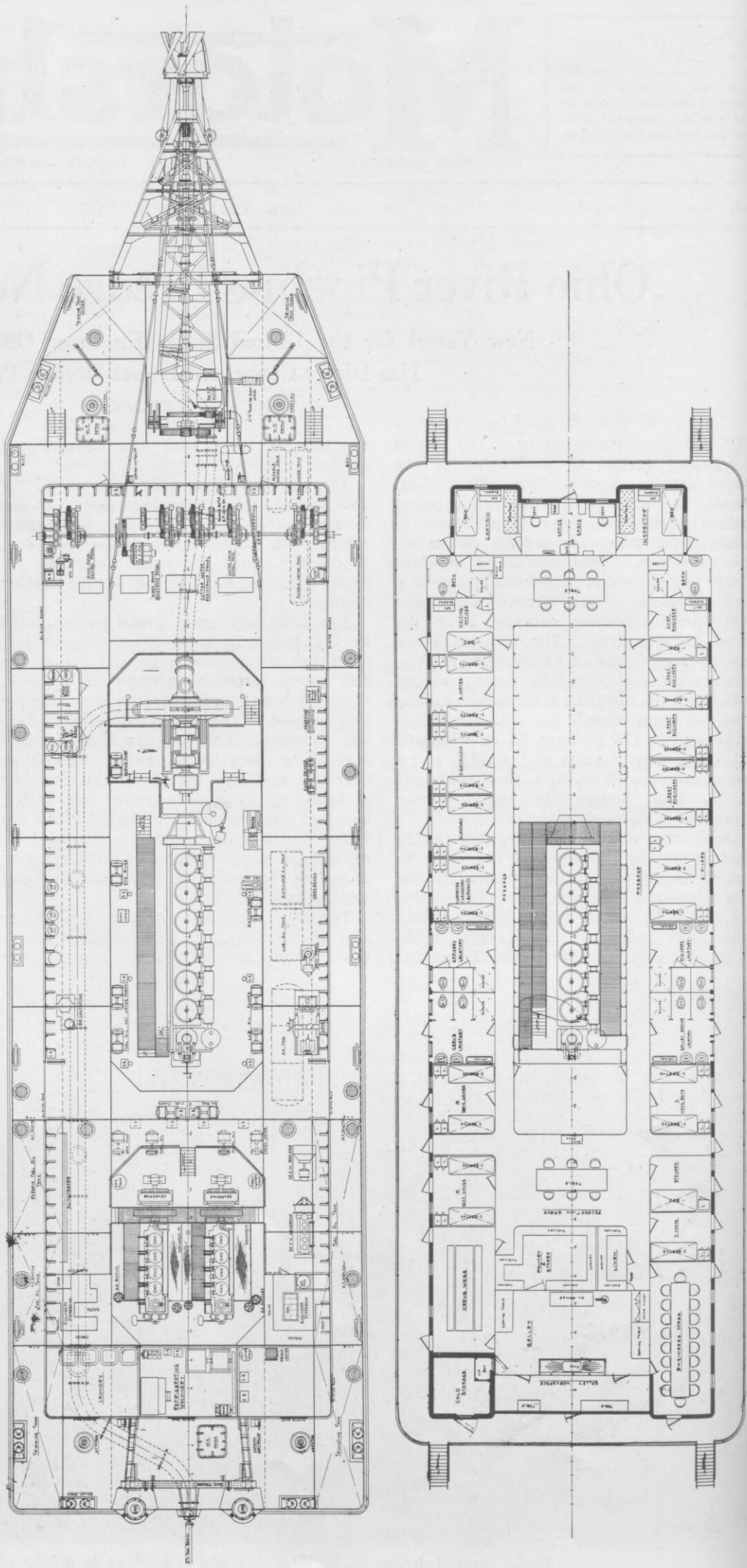
Type		*Gal. Min.*	*Hp.*
2	Lubricating oil pumps.....	60	3
2	Fuel oil pumps	50	5
2	Hot water pumps for fuel oil heating.....	60	2
1	Distillate transfer pump...	50	2
1	Service pump.....	75	5
1	Fresh water pump.....	75	3
1	Sanitary pump.....	75	3
1	Engine room bilge pump...	75	3
1	Hoisting and hauling gear pressure pump.....	40	3

CENTRIFUGAL PUMPS

Type		*Gal. Min.*	*Hp.*
2	Engine circulating water pumps	600	15
1	Fire and bilge pump.....	460	35
	Total		79

All rotary pumps are of the Northern Apparatus Company's type.

For supplying the starting air for the Diesel engine, a vertical two-stage air



Hold plan and main deck plan of Diesel engined dredge C. B. HARRIS, showing layout of machinery and dredging equipment

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Dredge C. B. HARRIS with suction ladder and spuds lifted

compressor, built by the Worthington Pump & Machinery Corporation, is installed. It is driven through a silent chain by a 20 hp. motor and is located in the engine room. This compressor has a piston displacement of 50 cu. ft. per min., and is capable of maintaining a pressure of 350 lb. per sq. in. continuously.

The dredge is equipped with a refrigerating plant. This installation in capable of maintaining a temperature of 40 deg. Fahr., in the cold-storage room and can produce 250 lb. of ice per day. It is motor driven and located in a separate compartment near the spud end of the dredge. The ice-making tank is in the same room where the refrigerating machine is installed, and the brine is circulated through the coils of the two cold storage rooms on the upper deck, located next to the galley.

A small auxiliary generator, of 25 kw. has been installed near the two auxiliary engines at the after end of the dredge, and this generator is intended for the purpose of supplying the current for the lighting system at night, in case the other generators are shut down and also for driving any of the auxiliaries when needed. This generator is operated by a gasoline engine and is of the G. E. Type GM-12. A 90-gal. gasoline tank is installed in a separate watertight compartment and equipped with filling line from the outside of the deck house.

On the starboard side along the main engine, a compact and complete machine shop is provided for, consisting of the following machine tools:

1—18/36-in. gap lathe.
1—18-in. shaper.
1—30-in. drill with sliding head.
1—2-wheel emery grinder.
1—drill grinder
1—hack saw to cut 5 in.
1—30-in. diameter grindstone.

All of these machines are arranged for individual motor drive, and a heavy oak work bench with the necessary drawers, machinist and pipe vises, also a large assortment of small tools, complete the equipment, which is intended to take care of any emergency that may have to be done on board this dredge.

The quarters on the upper deck consist of very plain staterooms in which the steel construction of the house and divisional bulkheads is exposed, with the exception of the outside wall, which is sheathed with beaver board, obtaining a simple panel effect.

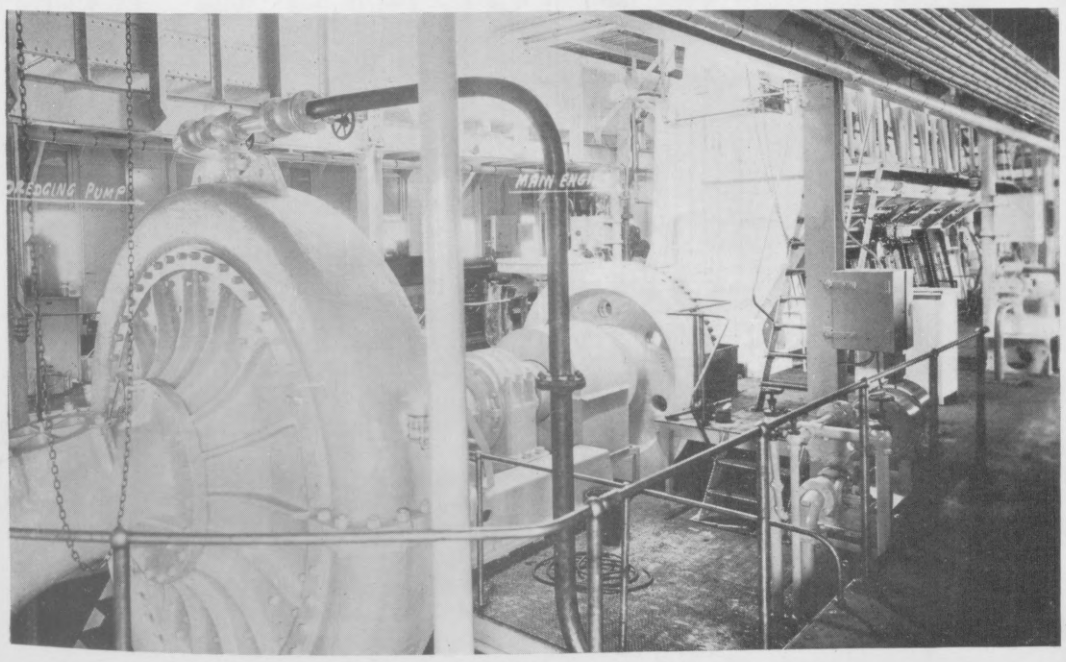
The forward end of the quarters is taken up by the cabins for the captain and inspector, with an office space in the center. The visiting officer and chief engineer each have a separate cabin directly behind these spaces. A bathroom with communicating doors to two of the staterooms give the two cabins on either side a combined bathroom. All other staterooms, with the exception of the steward's, are fitted with two berths each, providing quarters for the:

3 Mates	1 Launchman	6 Oilers
3 Electricians	1 Blacksmith	8 Deck hands
3 Lever men	6 Assistant engineers	2 Cooks
1 Carpenter		3 Mess boys

The officers' mess is arranged at the forward end of the two passages which lead on each side between the cabins and the engine casing, whereas the engineers' mess is on the after starboard side. In the center of the space between the engine inclosure and the galley a reception room is fitted up with tables and chairs for the comfort of the crew. A spacious galley workspace, screened at the after end, is located adjacent to the galley, and the pantry stores are accessible from the galley and located in a large storeroom immediately in front. Four lavatories, two on each side, are located about amidship, between the quarters for the officers and the crew and fitted with shower bath in each lavatory.

The accompanying reproductions of photographs and drawings are self-explanatory and it is believed that this dredge will attract considerable attention on account of the size and some unusual features incorporated in the design.

New river transportation projects are under consideration and further improvements of the Ohio River will, of course, help to make such projects materialize. Without doubt the U. S. Dredge C. B. HARRIS is going to share largely in the completion of the program mapped out in connection with the Ohio River improvements.



Machinery room of dredge C. B. HARRIS

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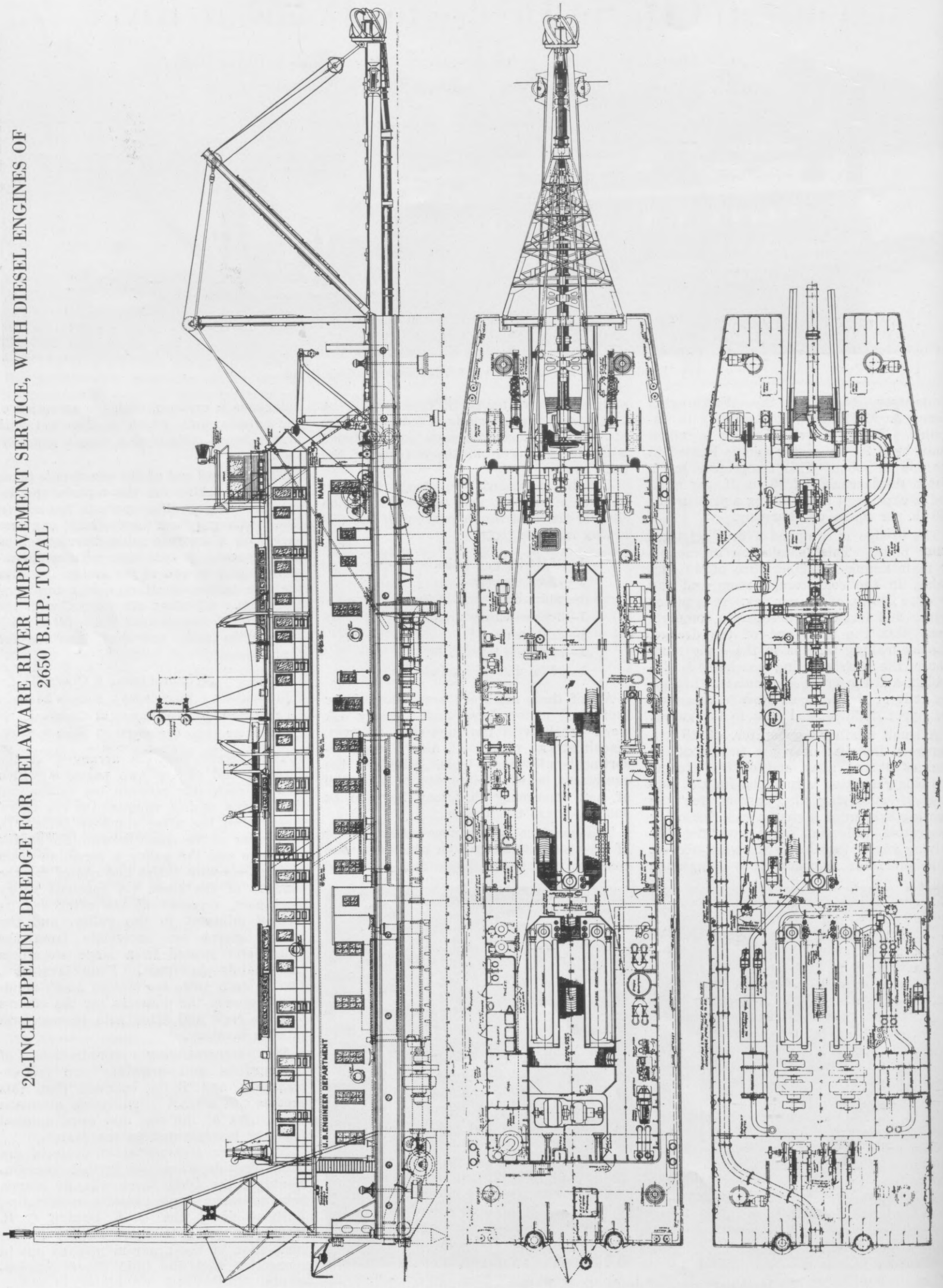
20-INCH PIPELINE DREDGE FOR DELAWARE RIVER IMPROVEMENT SERVICE, WITH DIESEL ENGINES OF 2650 B.HP. TOTAL



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Powerful Diesel Dredge for Delaware River

Latest Design by United States Engineer Office, Philadelphia, Has More Power Than Previous Machines Used in Similar Service

CONTINUING its policy of building the finest aggregate of dredging machinery to be found anywhere in the world, the War Department is going to build for Delaware River improvements, between Philadelphia and the sea, a 20-inch Diesel engined pipeline dredge. The announcement was first made in our columns two months ago, when a brief summary was given, which can now be expanded. The contract will probably be awarded early in April.

Following the practice first incorporated in the C. B. HARRIS, the new pipeline dredge for the Ohio River, the Philadelphia boat will have a centrifugal dredging pump direct-connected to a 1000 b.hp. Diesel engine. All the rest of the machinery is electrically operated, power being supplied by a pair of 750 b.hp. Diesel engines direct-connected to 500 kw. 230 v. compound-wound generators. The leading dimensions are:

Length molded	170 ft. 0 in.
Length overall	227 ft. 10½ in.
Breadth, molded	40 ft. 0 in.
Depth molded	13 ft. 0 in.
Mean draft loaded and with both spuds up	8 ft.

She is to be built of steel, under the rules of the American Bureau of Shipping. Bilge plates will be 7/16 in. throughout, with bottom, sides and ladder well plating of ⅜ in., bow plating of the same dimensions and transom plating ½ in. Longitudinal seams will be lapped and double riveted, and the butts in the bottom plating will be the same, but the butts in the side and bilge plating will be strapped and double riveted.

Floors are worked transversely from frame No. 8 to frame No. 68, spaced 30 in. heel to heel and are to be of 10 in. x 2.7 in. x 20 lb. channels, 4 in. x 3 in. x ⅜ in., 3 in. x 3 in. x ⅜ in. and 3 in. x 3 in. x 5/16 in. angles, with ⅜ in. lightened plates. Aft of frame No. 8 the floors will be worked longitudinally and spaced also 30 in. heel to heel. They will be built of 8 in. x 2.3 in. x 13.75 lb. and 10 in. x 2.7 in. x 20 lb. channels, 6 in. x 3½ in. x ⅜ in. and 3 in. x 3½ in. x ⅜ in. and 3 in. x 3 in. x ⅜ in. angles, with ⅜ in. lightened plates. Between frames Nos. 9 and 24, 26 and 43 and 45 deep floors will be fitted inboard of the longitudinal bulkheads to suit the engines and dredging pump. Outboard of the longitudinal bulkheads deep floors will be fitted at frames Nos. 11, 13, 15, 17, 19, 24, 27, 29, 31, 33, 35, 37, 39, 41 and 44 on both sides, and at frames Nos. 22 and 23 on the starboard side and on frame No. 21 on the portside. Floors at frames No. 21 on starboard side and No. 23 on port side will be oil tight outboard of the longitudinal bulkhead, and floors at frames Nos. 22 and 24 will be water tight between inboard longitudinal engine girders.

The side frames will be of 6 in. x 3½ in. x ⅜ in. angles spaced 30 in. heel to heel, lapped on to deep floors and joined to deck and floor beams by ⅜ in. plate brackets.

Engine girders are continuous inboard on frames Nos. 8 to 25 and from frames Nos. 25 to 44, outboard from frame Nos. 8 to 25 only. These girders will be of ⅜ in. lightened plates connected to the shell by 3 in. x 3 in. x ⅜ in. angles. Top plates in way of the engines and dredging pump will be ⅝ in. connected to the girder by 5 in. x 3½ in. x 7/16 in. angles. Transverse floors will be connected to the girders by 3 in. x 3 in. x 5/16 in. angles. Girders between frames Nos. 8 and 25 will be 3 ft. high, between frames Nos. 25 and 38 will be 5 ft. high, and between frames Nos. 39 and 46 will be 4 ft. 10 in. high to suit the engine and dredging pump.

Water tight bulkheads will be located on frames Nos. 8, 50, 54 and 57, constructed of 5/16 in. plate, except the latter, which will be of ⅜ in. plate. Oil tight and water tight bulkheads will be fitted on frames Nos. 25, 36 and 44, the former constructed of 5/16 in. plate, and the two latter of ⅜ in. plate in way of the fuel oil tank. A longitudinal bulkhead is fitted 10 ft. from the center line between frames Nos. 8 and 57 on port and starboard sides, water tight and oil tight only between a few frames as required. An oil tight longitudinal bulkhead extends between frames Nos. 25 and 36 on the port side, located 14 ft. 6 in. from the center line, and between frames Nos. 25 and 44 on the starboard side located 17 ft. 6 in. from the center line. This plating is ⅜ in. except between frames Nos. 8 and 25 and between frames Nos. 50 and 57 where the center strake is to be of 5/16 in. The main deck will be laid flat and of ⅜ in. plating aft of frame No. 65½ and ½ in. forward of that.

In this dredge the two spuds will be 75 ft. long, with an outside diameter of 34 in., built of ¾ in. steel plates with double riveted seam straps and treble riveted butt straps, each spud being stiffened with six continuous 4 in. x 3 in. x ½ in. angles, and by seven ⅜ in. plate diaphragms with 3 in. x 3 in. x ⅜ in. angle clips. At the top of the spud a single 30 in. sheave will be fitted to take a 1 in. diameter wire hoisting rope. For these spuds the hoisting frames will consist of two columns extending to 68 ft. 6 in. above the main deck and located 4 ft. 8¾ in. off the center line on port and starboard sides. Each column will be built of two 10 in. x 2.6 in. x 15.3 lb. channels, supported by a backstay of two 8 in. x 2.26 in. x 11.5 lb. channels, all held together by means of tie plates and lattice bars.

The A frame for the dredging ladder will be 42 ft. 3 in. long, consisting of two inclined columns each constructed of one 15 in. x ⅜ in. web plate, four 5 in. x 3½ x ⅜ in. angle, and two 11 in. x ⅜ in. face plates. The columns are pin connected at their lower ends to cast steel bases and are connected together by double 5 in. x 3½ in. x ⅜ in. horizontal angle bars and 3 in. x 3 in. x ⅜ in. double diagonal angle bar bracings. On top of this frame there will be fitted five 30 in. sheaves to take 1 in. diameter hoisting wire for the latter.

The gallows frame is to be 31 ft. 3 in. long, consisting of two latticed inclined columns, each constructed of two 12 in. x 3.047 in. x 25 lb. channels, and 2¼ in. x 5/16 in. single lattice bars. The columns are pin connected at the lower ends to cast steel bases and braced strongly together. On top of the frame are two cast steel seats for the stayrods from the A frame, which are to be 3¼ in. diameter, and for the stayrods connected to the hull of the dredge which are to be 3½ in. diameter.

As the main power equipment there are three Diesel engines, one direct connected to the dredging pump, and two direct connected to d.c. generators. The pump is of the centrifugal type with a self-aligning roller type bearing at the pump-end of the shaft and a deep groove self-aligning ball type thrust and radio bearing between the pump and the main engine. A flexible coupling will be fitted capable of transmitting 1000 b.h.p. at rated engine speed.

An unusually extensive auxiliary operating equipment is to be added to the engines, including pyrometers. Crane equipment is also provided above the engines. A double track 5-ton crane will be installed between frames Nos. 54 and 57, whilst a 2-ton hoisting block running on a single I-beam will be fitted over the Diesel engine on each side between frames Nos. 11 and 26 and also on the center line in the engine casing between frames Nos. 26 and 38

An exhaust gas steam generator is to be used for providing steam for general purposes on board. In addition to the main engines there are to be also a 120 b.hp. oil-engine direct-connected to a d.c. generator of 85 kw. and a similar engine connected to a 10 kw. lighting generator.

For all the lubricating oil used on board there is to be a tank with a capacity of 1200 gal. For the clean oil there will be another tank of 400 gal. capacity and in the engine room there will be a service tank of 100 gal. capacity, in addition to three auxiliary tanks. Connected with this system will be two motor driven oil purifiers of the centrifugal type.

In the fuel oil system no provision is made for centrifuging, but it may be noted that fuel oil heaters are installed. These heaters have a capacity for heating 600 gal. of 16 deg. Beaumé oil from 60 deg. F. to 110 deg. F. in one hour when supplied with steam at a pressure of 15 lb. per square inch.

An auxiliary high pressure air compressor is provided and there is also a low pressure air compressor, both to be electrically driven. In addition there are numerous pumps and other auxiliaries.

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A System for Centrifuging Boiler-Fuel Oil

As Planned for the 3400 b.hp. Diesel Engine Installation of the Port of Portland's New Dredge

IN working out his plans for the Port of Portland's powerful new Diesel-electric dredge CLACKAMAS, Mr. Polhemus, the general manager, who already has considerable pioneering to his credit, has planned to use boiler-fuel oil in the engines in place of the regular Diesel oil. The mechanical arrangement of the fuel system having been finally determined after considerable investigation and thought, Mr. Polhemus has released the details of it for publication, in order that others may have the benefit of the intensive study he has conducted.

With a main power plant consisting of two 900 b.hp. and two 800 b.hp. McIntosh & Seymour 4-cycle trunk-piston engines driving independent d.c. 500-volt generators, the fuel consumption is estimated at about 40 bbl. per 8-hour shift. As the dredge is to be operated continuously night and day with three 8-hour watches, it was decided to treat the fuel in 40 bbl. lots, good for one full shift's operation.

For this purpose a comprehensive tank lay-out has been planned in conjunction with a large centrifuge. The procedure will be to pump fuel-oil from the general fuel bunkers of the dredge into the upper tanks of the centrifuging system. These are the tanks marked Nos. 1 and 3 in the drawing. There the oil is heated to about 140 deg. Fahr. by means of hot water coils inside the tanks.

About two hours will be required to raise the fuel to that temperature, during which process the emulsified water and oil should be broken up, allowing most of the water to settle in the bottom of the tanks before the heated oil is run through the centrifuge to the two day tanks marked Nos. 4 and 6.

The day tanks, which are fitted with hot water coils to keep the purified fuel oil at 140 deg. Fahr. or any other temperature desired, are set high up in the dredge so that there will be a head on the suction side of the engine fuel pumps. In each of the upper or heating tanks there is a surface of 300 sq. ft. of hot water coils, while in each of the lower or day tanks there is 150 sq. ft. Hot water will be circulated at about 200 deg. Fahr. through these coils. The same hot water is used to heat the fuel oil about the suction intakes in the various bunker tanks, and is further employed for heating in the living quarters of the dredge. The hot water system is a closed one. The heat is taken from the engine mufflers by means of heating coils.

While the engines are running on one of the day tanks, and the tank above it is being simultaneously filled and heated, the other upper tankful of oil, having already been heated and settled for eight hours, is being run into the other day tank, after passing through the centrifuge, and is left there for use on the next engine shift. The day tanks are carefully calibrated and gauged so that the fuel consumption can be accurately determined at all times.

As the upper tanks are set above the day tanks, the fuel oil passes through the centrifuge without the aid of pumps. A No. 6 Sharples mechanical centrifuge is used, having a rated capacity sufficient to

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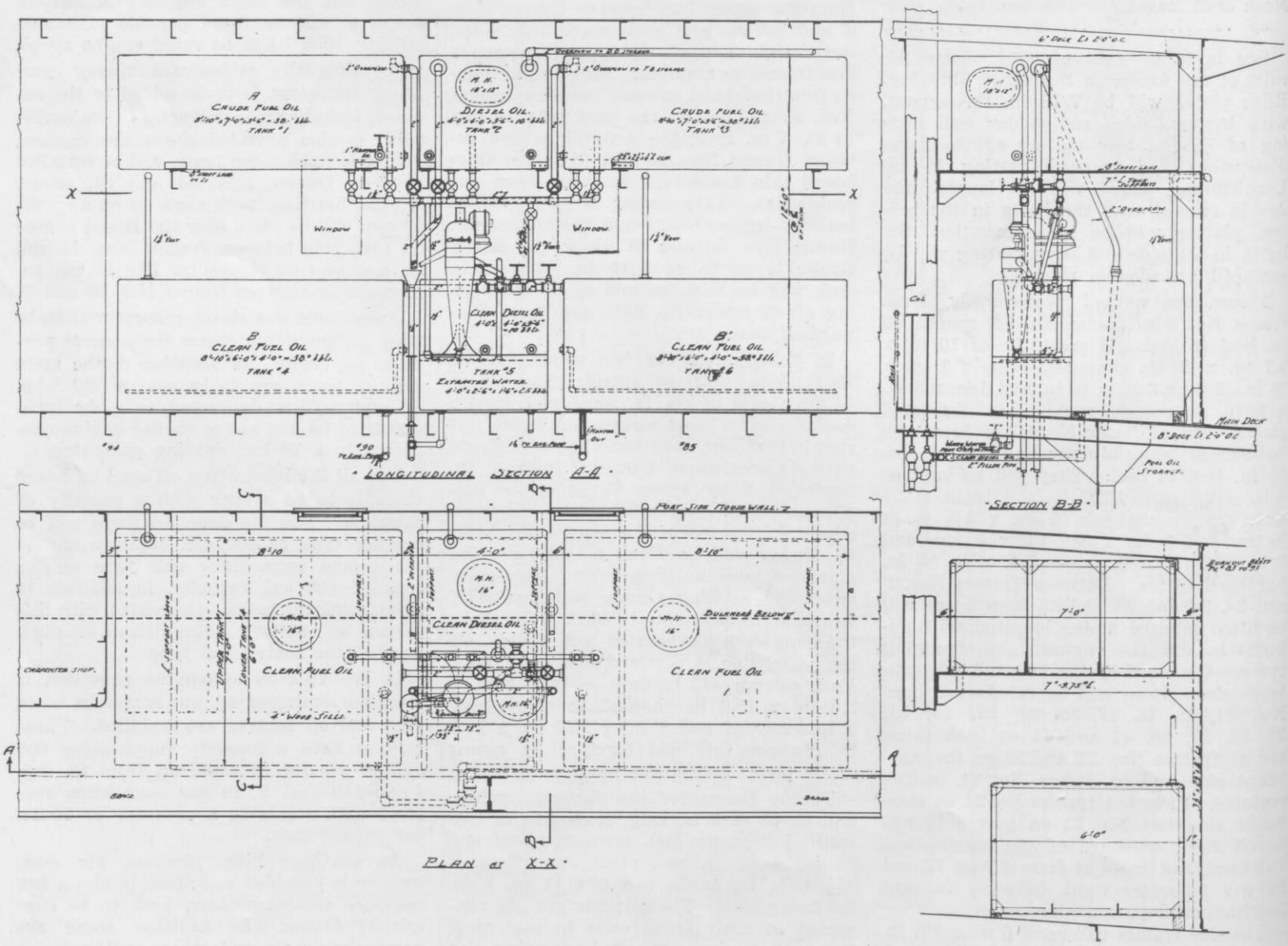

Arrangement of the Installation for Centrifuging Boiler-Fuel Oil on the Powerful Diesel-Driven Dredge CLACKAMAS Now Under Construction

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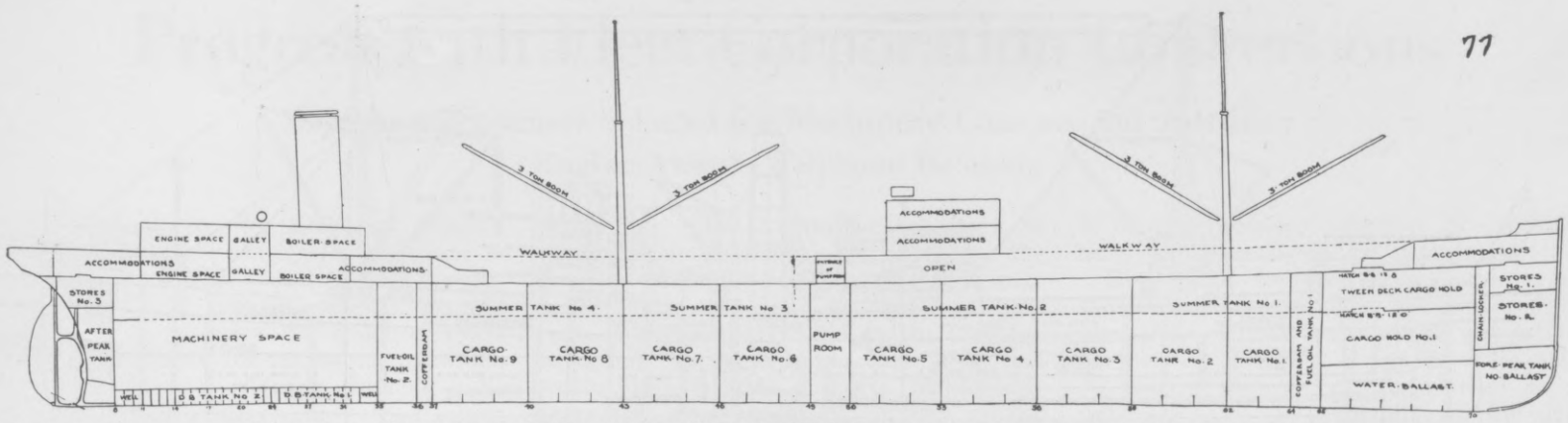

Inboard profile of the tankship purchased by the Gulf Refining Co. for conversion to Diesel power

Gulf Refining Co. to Convert Tanker Miskianza

Will Install 2500 s.hp. Sun-Doxford Diesel Engine in 9971 Tons dw. Bulk Oil Carrier Purchased from the Shipping Board

THE Gulf Refining Company, which has been studying problems of Diesel power for about ten years, is prepared to give this type of power a thorough try-out. It has just purchased a steam tanker from the Shipping Board for conversion to Diesel power. The choice has fallen upon the 9971 tons dw. vessel MISKIANZA, built in 1920 by the Moore Shipbuilding Co., San Francisco. An order for her conversion has been placed with the Sun Shipbuilding Co. of Chester, which will take out the old machinery and replace it with a 4-cylinder, 2-cycle type opposed-piston Diesel engine of the Sun-Doxford design, developing 2500 s.hp. at 78 r.p.m.

All the engine-room auxiliaries will be of Diesel-electric type, but one boiler will be retained for the purpose of heating or steaming cargo tanks, etc. Plans are being prepared to utilize the exhaust gases for power purposes, although at the time of writing the actual system to be adopted has not yet been decided.

While the oil-engines and generator sets to be used as auxiliary power have not yet been definitely fixed, plans call for the use of two Ingersoll-Rand oil-engines of 75 b.hp. each.

The following are the general dimensions of the MISKIANZA:

Deadweight capacity	9971 tons
Length o. a.	438 ft. 5 in.
Length, b. p.	425 ft. 0 in.
Breadth, md.	57 ft. 0 in.
Depth, md.	32 ft. 0 in.

At the same time that the Gulf Refining Company purchased the MISKIANZA, the Associated Oil Co. of San Francisco bought the 10,250 tons dw. steam tanker TULSAGAS from the Shipping Board for $310,000 cash, together with an agreement that they will install a reciprocating steam engine in place of the present turbine equipment of the ship. The purchase price of the MISKIANZA was $150,000, the buyer agreeing to install Diesel engines. When these two vessels are in service it will be very interesting to obtain a comparison of the operating costs and figure out the overhead on the basis of first-cost with a view to ascertaining which is the most satisfactory plan for shipowners to follow when purchasing vessels from the Board.

The total capacity of MISKIANZA'S tanks is 10,276 tons, distributed as follows:

Tank	*Cu. Ft.*	*Bbl.*	*Tons of 37.5 Cu. Ft.*
No. 1 port and starboard.....	31,390	5,590	837.1
No. 2 port and starboard.....	46,840	8,342	1,249.1
No. 3 port and starboard.....	45,612	8,124	1,216.3
No. 4 port and starboard.....	44,834	7,986	1,195.6
No. 5 port and starboard.....	42,638	7,594	1,137.0
No. 6 port and starboard.....	42,458	7,562	1,132.2
No. 7 port and starboard.....	44,092	7,854	1,175.8
No. 8 port and starboard.....	44,012	7,840	1,173.6
No. 9 port and starboard.....	43,476	7,744	1,159.4
Totals	385,352	68,636	10,276.1

A System for Centrifuging Boiler Fuel Oil

(*Continued from previous page*)

run a 40 bbl. lot of oil through in about seven hours. A spare rotor is carried which is changed every watch.

The problem being a new one to Mr. Polhemus and his staff, they had some interesting engineering features to work out, such as how much radiation could be obtained from the hot water system, the temperatures to which the fuel oil should be heated, and how long the initial settling process in the upper tanks should be.

In these considerations a helpful guide was the experience already had in burning fuel oil of the ordinary steam boiler grade in the 525 b.hp. Pacific-Werkspoor engine of the 15-inch Diesel-electric dredge TEXAS during the past year and a half.

In that installation two day tanks of 8-hour capacity each were used, the fuel oil was heated by hot water coils and the excess water merely drained off after eight hours' heating and settling before being fed to the Diesel engine. After better than a year's constant operation on the 24-hour a day schedule this engine was taken down, and little or no wear was evident on the cylinder walls or pistons. No undue trouble had been experienced with valves, through either the sulphur contents or asphalt contents of the fuel oil, and the operating engineers preferred the straight boiler fuel to the special Diesel oil. In fact the way the engine was tuned up it ran better on the fuel oil than on the Diesel oil and the engine exhaust was cleaner. The boiler fuel also seemed to have better lubricating qualities.

By introducing the centrifuging system on the dredge CLACKAMAS and burning straight commercial boiler fuel oil as supplied by the California oil companies, the Port of Portland expects to show a yearly saving of between $15,000 and $20,000. The mechanical centrifuge will extract the water that does not settle and the suspended impurities such as sand, but, of course, is not expected to eliminate any of the sulphur or asphalt contents that the fuel may have.

As seen in the drawing there is a second, but smaller tank system, in the installation. This is for handling the oil fuel used to fill the fuel pipe lines of the main engines just before shutting down and which is used also to operate the 110 b.hp. Atlas-Imperial solid injection type of engine that drives the auxiliary unit on Sundays and holidays and turns a 75 kw. generator when the main power plant of the dredge is shut down.

Mr. Polhemus adds that he obtained the collaboration of representatives of the engine builders, centrifuge makers and operators of the Long-Bell Lumber Company's Diesel-electric dredge TEXAS in reviewing the problems that had to be met and in determining how to cope with them. Success would thus appear to be assured, and its realization will encourage many other operators.

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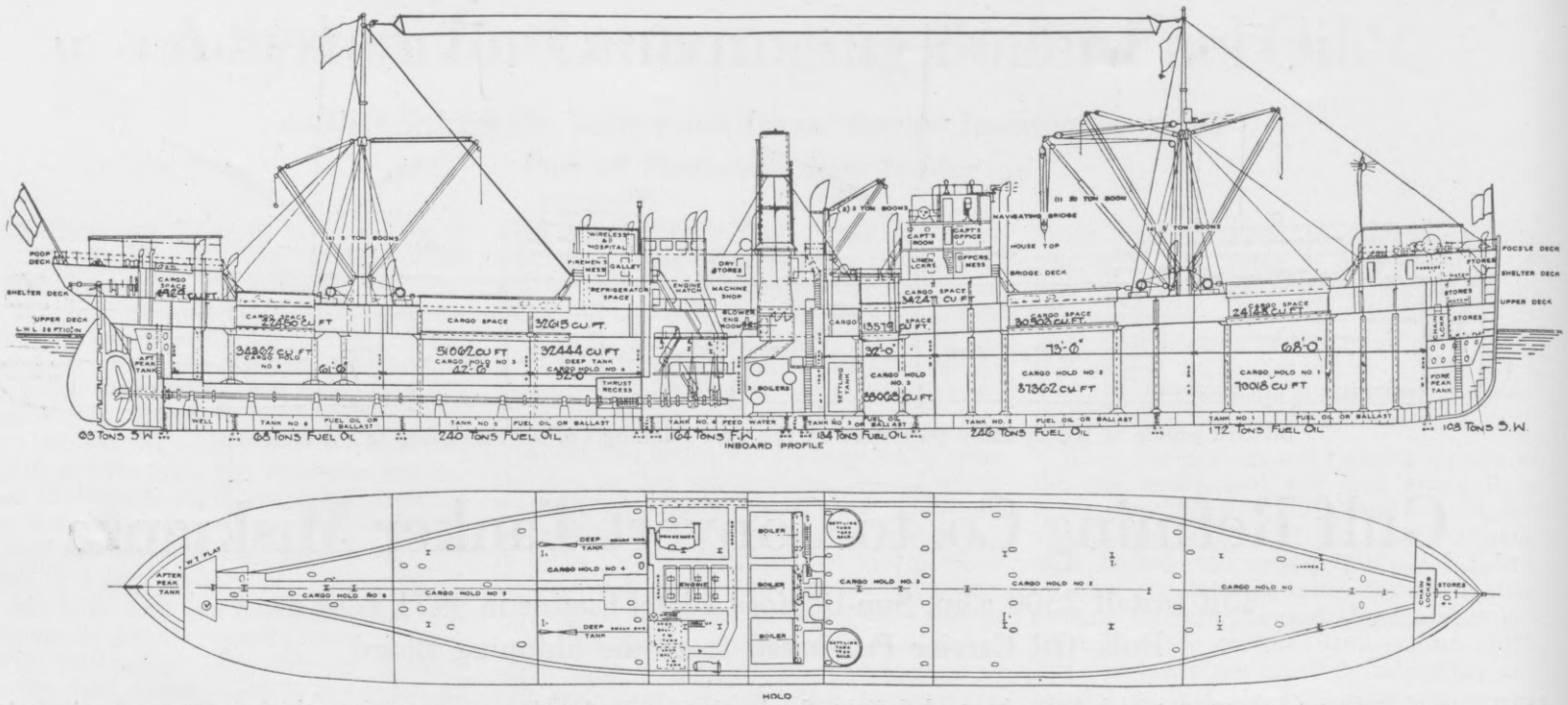
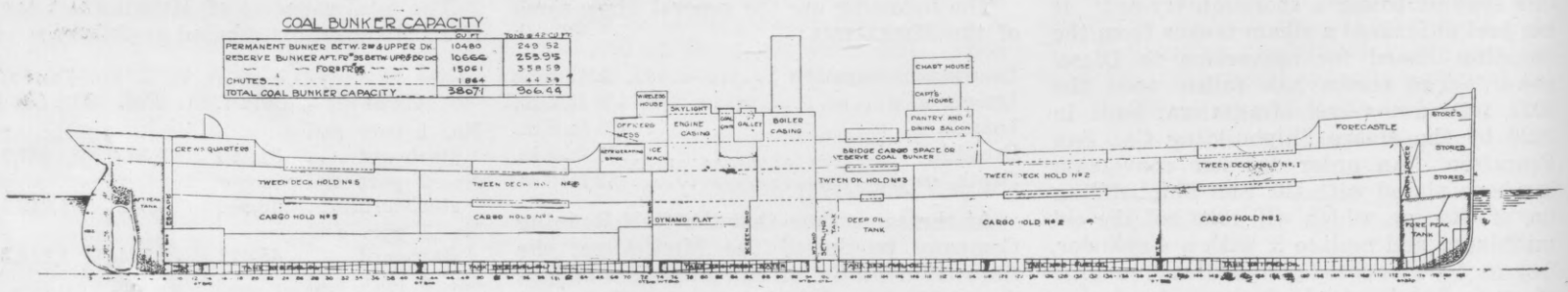
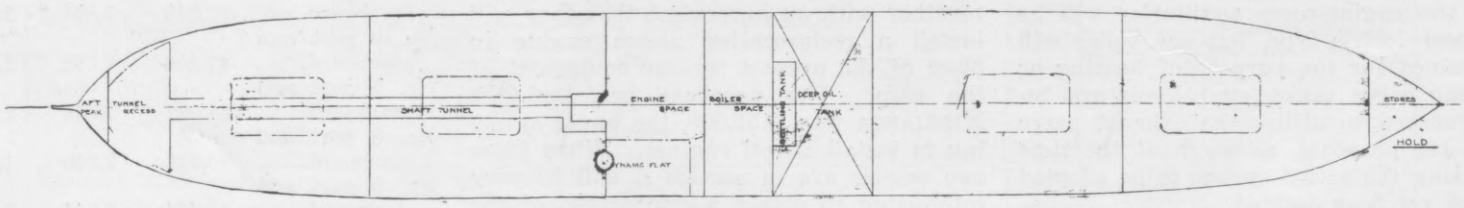
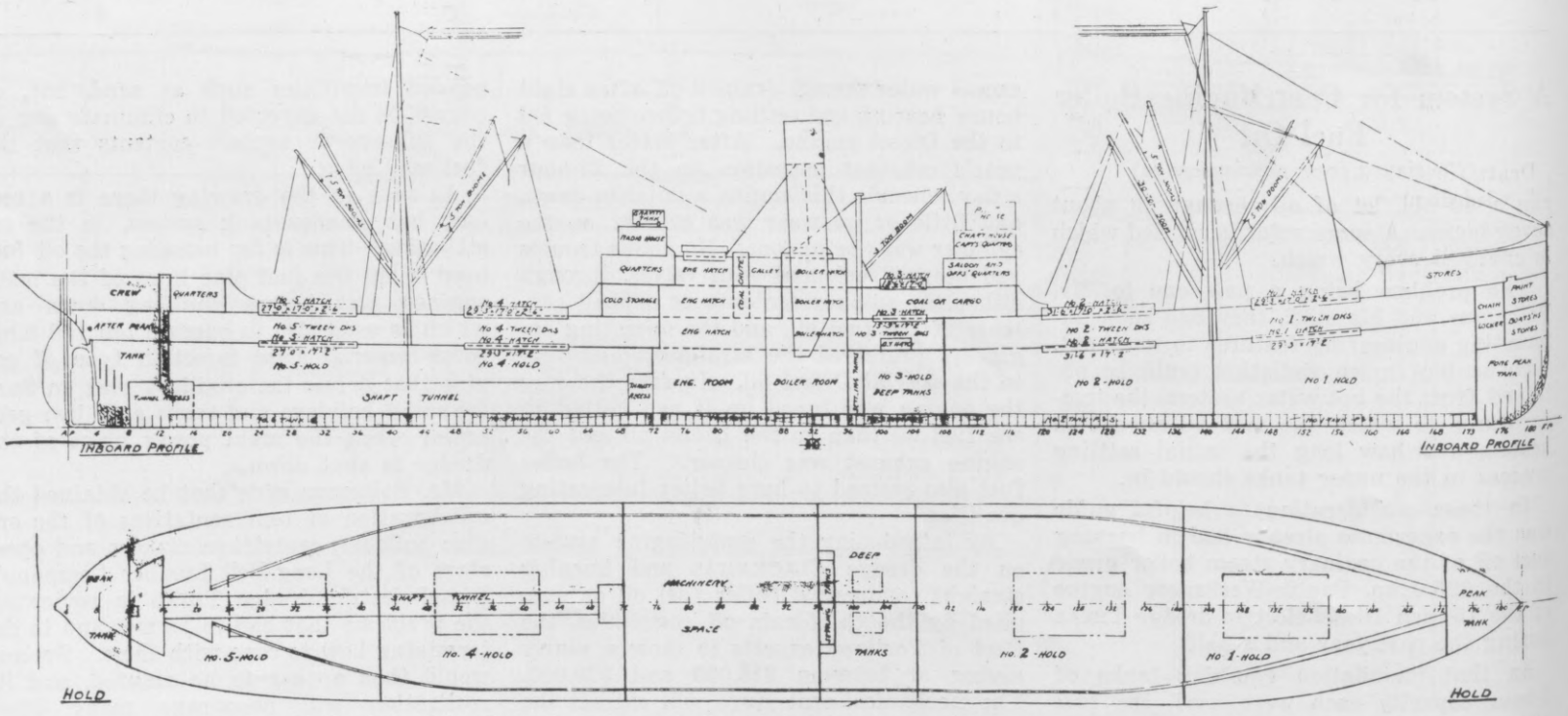


Eight vessels of the Oscar Daniels class, shown above, have been selected for conversion to motor power


COAL BUNKER CAPACITY
PERMANENT BUNKER BETW. 2W & UPPER DK. 10480 240.52
RESERVE BUNKER AFT. FR 35 BETW UPR & PR DKS 10666 255.93
FORE PEAK 15061 358.59
CHUTES 1864 44.39
TOTAL COAL BUNKER CAPACITY 38071 906.44



Of this class of vessel, the Los Angeles type, three ships will be converted under the first part of the Government's program



Two ships of the Columbia River class, very similar to the Los Angeles class, have been chosen for Diesel power

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Progress with Fleet Corporation Conversions

Majority of Steamers Selected for Machinery Changes and Auxiliary Engine Awards Will Soon Be Made

PROGRESS on the Shipping Board's initial steamer conversion program is steadily proceeding. The Fleet Corporation is carefully planning each step, and with a determination to produce craft eminently suitable for the routes on which they will operate. So far thirteen hulls of three different classes have been selected, and it is anticipated that in a few weeks the remaining five vessels will also have been chosen.

The boats selected to date comprise two of the Columbia River class, three of the Los Angeles type, and eight of the Oscar Daniels build.

All thirteen vessels are of the single-screw type with midship engine-rooms, and may be considered seaworthy craft in good condition. In making the selection the Fleet Corporation has had before it the fact that the converted vessels must immediately be placed in service under the terms of Section 12 of the Ship Conversion Act of 1924 unless a binding sale contract was made prior to re-conditioning. The Fleet Corporation is faced with the problem that under today's ocean transportation conditions the resultant motorships cannot be placed in service just anywhere, but to an extent must replace certain steamers now operating on long distance voyages, some of which in any event are due to be withdrawn for overhaul in a year to eighteen months' time.

Naturally the Fleet Corporation desires the best and most suitable hulls for the purpose and has had no intention of selecting craft laid-up for a long period and which may need a large expenditure to render them efficient. Then again, the question of engine-room space, size of propeller shaft, and other but less important reasons, all carried weight in the decision. It is only to be expected that opposition would arise in some quarters, particularly from the boiler industry, on the ground that the Fleet Corporation should convert some of the lame ducks now laid up rather than take steamers out of service to convert them into more efficient freight carriers. A little consideration clearly indicates that such an attitude is a narrow one and is mainly raised by reason of the conversion program directly affecting the businesses of individual concerns.

When all facts are borne in mind and carefully weighed, the action of the Board is more readily understandable. Even if some of the steamers now in service were not withdrawn at this time, they would have to be taken off their routes when the second part of the Board's program commences. But continuation of Federal conversion, as well as similar action by private shipowners, largely depends upon the success of the first eighteen ships, and the Fleet Corporation does not consider it a wise policy to risk any possibility of inefficient operation by the selection of laid-up hulls which are not the best suited for 100 per cent success. It must not be forgotten that the machinery going in these vessels is costing in the neighborhood of $350,000 per ship, and as these are the first conversions to be carried-out by the Board, and consequently are in the nature of sample vessels, selection of doubtful hulls would hardly be an economy.

Bids from fourteen manufacturers of auxiliary Diesel engines were opened on March 6. Prices were asked on a total of forty-nine, 75-kw. d.c. generating sets. No useful purpose can be served by a publication of the summary of the data at this time. Several proposals were not in order and consequently could not be given serious consideration; others were submitted which were not properly in line with the specifications and have been rejected. When the awards are made a tabulation of the data will have more constructive results.

On the whole, bidding on the auxiliary engines was considered successful by the Fleet Corporation, because the Board was able to get from a number of leading manufacturers a specification article at a price considered reasonable. Captain R. D. Gatewood and his department have made a thorough analysis of the bids and their recommendations have been sent to Washington, and doubtless action will be taken regarding the awards shortly after this issue is published.

Bearing in mind the fact that a number of responsible bidders quoted on an article which meets the specifications, it looks as if the Board will be able to award a considerable number of the engines to the low bidders. In cases where prices are reasonable and other things are considered it is natural also that some of the engines should be awarded to the builders of main engines already on order.

Principal Measurements of the Three Classes of Vessels Already Selected for Conversion

COLUMBIA RIVER CLASS

Ships selected—WEST HARTS and WEST HARTLAND.

Design No.	1013
D. W. tonnage	8800 tons
Cubic capacity (grain)	445,084 cu. ft.
Length, o.a.	423 ft. 9 in.
Length, b.p.	410 ft. 5½ in.
Breadth	54 ft.
Depth	29 ft. 9 in.
Draft	24 ft.
Loaded displacement	12,000 tons
Designed power	2500 s.hp.
Designed speed	10½ knots
Propeller shaft diameter	14⅛ in.

LOS ANGELES CLASS

Ships selected—CROWN CITY, WEST HONAKER and WEST CUSSETA.

Design No.	1013
D. W. tonnage	8400 tons
Cubic capacity (grain)	461,178 cu. ft.
Length, o.a.	423 ft. 9 in.
Length, b.p.	410 ft. 5½ in.
Breadth	54 ft. 10 in.
Depth	29 ft. 9 in.
Draft	23 ft. 11¼ in.
Loaded displacement	12,086 tons
Designed power	3000 s.hp.
Designed speed	11 knots
Propeller shaft diameter	14½ in.

OSCAR DANIELS CLASS

Ships selected — YOMACHICHI, WILSCOX, TAMPA, SEMINOLE, UNICOI, SAIVOLSIA, CITY OF BAYVILLE and CITY OF DAHLHART.

Design No.	1027
D. W. tonnage	9300 tons
Cubic capacity (grain)	515,888 cu. ft.
Length, o.a.	416 ft.
Length, b.p.	402 ft.
Breadth	54 ft.
Depth	33 ft. 9 in.
Draft	26 ft. 10¾ in.
Loaded displacement	12,910 tons
Designed power	2200 s.hp.
Designed speed	10.5 knots
Propeller shaft diameter	14½ in.

A new Diesel yacht is being built for Capt. J. K. L. Ross of Montreal, Canada, one of the directors of the C. P. R. and widely known as the owner of the famous race horse Sir Barton. Mr. Ross has been a yachtsman for many years. His new yacht will be about 100 ft. in length overall, 19 ft. beam and 6 ft. draft and will be equipped with two 6-cylinder 150-hp. Winton engines which are expected to drive her at 12 knots. The designs were prepared by Tams & King.

Ogden Mills purchased the motor yacht ALCALDA, which was chartered to him last summer, and is now having her converted from plans and specifications by Tams & King into a twin screw Diesel yacht. This yacht is 105 ft. overall, 16 ft. beam and 5 ft. draft. When she was originally built she was equipped with two 300 hp. gasoline motors. They are being replaced by two 150 hp. reversible Winton Diesel motors, which will reduce the operating cost by about $10 an hour.

RIPPLE, the twin screw yacht built for C. N. Leonard, has been sold to Thomas L. Chadbourne of New York. She was built in 1923 by Krupp's at Kiel and is a twin-screw 300-ton boat with two Krupp engines of 250 hp. each. She came over to this country under her own power and subsequently made a cruise to South America. Recently she has crossed the Atlantic again, making the passage from New York to Gibraltar in 15 days. Her new owner will bring her back in the summer.

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Canada Selects Motorships for Subsidy Plan

Ten Diesel Vessels to Be Operated in North Atlantic Trade If Parliament Ratifies Low-Petersen Agreement

RELIANCE is placed entirely on motorships to attain the ends sought by the Canadian Government in its plan for a subsidized North Atlantic service that will satisfy the cry of the Canadian farmers for cheaper ocean transportation.

This appears from the recent publication of the so-called Petersen agreement signed last December by Mr. Low, Minister of Trade and Commerce.

If the Dominion Parliament ratifies the contract, the subsidized service would have to be maintained by 10 single screw motorvessels of about 5000 tons gross each, driven by Harland & Wolff Diesel engines of 1850 s.hp. each and capable of maintaining about 11 knots speed. Two of these ships would have to be in service by Aug. 1 of this year and all of them by June 11 next year. Meantime the service would be started with suitable chartered steamers, not less than six to be on the schedule by June 15 of this year.

Exactly what the legal responsibilities of the parties are can best be gleaned from the terms of the agreement itself, which is set out in full hereunder:

(1) That the contractor shall and will either personally or by a company which he may form (and it is hereby understood and agreed that this contract may be executed, performed and fulfilled by a duly incorporated company formed or to be formed by or at the instance of the contractor, which company when formed shall be accepted by His Majesty instead of and in lieu of and in substitution for the contractor, and which shall be hereinafter referred to as the "shipping company") :—

(*a*) Build, equip, provide, establish and during this contract, continue and in the manner hereinafter mentioned, operate a regular shipping service between such points in Great Britain or Ireland or the continent of Europe and Canadian ports as may be designated by the Minister of Trade and Commerce for Canada or his representatives from time to time.

(*b*) Establish and continue and maintain and operate as hereinafter set out the shipping service hereinbefore referred to free from the control of and independent of the Atlantic shipping combine or conference or any other combine or conference now in existence or which may be operated or established upon the Atlantic Ocean at any time in the future without the sanction of the government of Canada.

(*c*) That such service shall be put in effect and maintained with ten ships as hereinbefore referred to, and these shall be built and placed in operation with all possible despatch and shall be fully completed and put in service within eighteen months from the date of this agreement (at least two of the same to be placed in service by 1st August, 1925), and pending completion of the same other ships of modern construction properly equipped and of approximately the same tonnage shall be substituted and operated in the said service until the fleet in question is completely constructed and in operation, the said service to commence with at least six ships not later than 15th June, 1925.

(*d*) That the control and regulation of the rates to be charged by the contractor shall rest with the government of Canada, and that the government shall fix such transportation rates on all commodities from time to time.

(*e*) Make proper provision for the carriage of perishable and partly perishable produce by means of cold storage facilities.

(*f*) Carry emigrants from Great Britain or the continent of Europe at rates to be fixed by the government, and carry all Canadian mails free, if and when the parties hereto agree upon the establishment of a passenger service or other steamers suitable for the carriage of emigrants in conjunction with and under the control of the shipping company.

(2) And His Majesty . . . shall well and truly pay or cause to be paid to the contractor, his heirs, administrators and assigns during the continuance of this contract an annual subsidy of two hundred and seventy-five thousand pounds (£275,000) from and out of the consolidated fund or other proper moneys of the Dominion of Canada, the same to be paid and payable to the contractor at . . .

(3) And it is mutually covenanted and agreed between the parties hereto that the subsidy payable to the contractor in the preceding paragraphs is based on the operating and maintenance of a service of ten ships, and the subsidy aforementioned shall be increased or reduced as the case may be in proportion to the number of ships operated and maintained by the contractor in the service aforesaid from time to time, and such increase or reduction of subsidy shall be added to or deducted from the monthly payments hereinbefore provided for in paragraph (2).

(4) And it is mutually covenanted and agreed between the parties hereto that the shipping services to be established by the contractor herein will cooperate and work loyally with the Canadian National Railways system and the fleet of ships operated by that system and under its control and known as the Canadian Government Merchant Marine.

(5) And it is mutually covenanted and agreed between the parties hereto that should any of the ships in the service aforesaid be lost or disabled the contractor shall have the privilege of substituting a modern ship of approximately the same tonnage for the ship lost or disabled until the same is replaced or repaired and put in service again as the case may be.

(6) And it is mutually covenanted and agreed between the parties hereto that on receiving at least three months' notice from the Minister herein the contractor shall arrange to place additional ships up to the number of ten in this service on an increase in subsidy in proportion to the number of ships actually placed in service; the subsidy referred to in paragraph (2) being based on a service of ten ships; and the additional service to be put in effect within twelve months from the date of the receipt by the contractor of the notice aforesaid.

(7) The contractor shall not convey or permit to be conveyed in any steamer while employed in this service any nitro-glycerine or any other articles which in the opinion of the Minister shall be considered dangerous.

(8) If at any time after the expiration of the eighteen months mentioned in paragraph 1 (*c*) the contractor should fail to carry on the said service with the full complement of ten ships the Minister may by notice in writing call upon him to supply the deficiency within a delay of three months, and in default of the contractor's doing so within such delay the Minister may terminate the contract by notice in writing.

(9) And the contractor covenants and agrees that the vessels in the said service shall prosecute all voyages made by them with all possible speed and despatch.

(10) The Minister shall have the privilege at his option of entering into contracts for like purposes with any other contractor or contractors.

(11) And it is mutually covenanted and agreed between the parties hereto that this contract is to continue in force for the full period of ten years from the time when the first service under the terms hereof is commenced; and the same may be extended for a further period of time and under such terms and conditions as the parties hereto mutually agree upon.

(12) Except to the company to be formed by the contractor as hereinbefore provided for to whom this contract may be assigned no other assignment thereof, either by the contractor or by the company, is to be or shall be made, nor any right nor interest therein, without the consent in writing of the Minister having been first obtained.

SCHEDULE "A"

Particulars (without guarantee) of two corrugated open shelter-deck motor-vessels, with topgallant fo'castle and cruiser stern building by Harland and Wolff, Ltd., at Henderson's, Glasgow:—

Class 100A1 and B.V.; length overall, 416 ft.; B.P., 400 ft.; breadth moulded, 55 ft. 6 in.; corrugation, 59 ft.; depth, 28 ft. 4 in.; S.D., 37 ft. 10 in.; deadweight (including bunkers), 9000 tons on 25 ft.; cubic capacity (about), 580,000 grain; tonnage (approximately), 5000 gross, 3500 net; capacity of oil bunkers, 750/1000 tons; height of 'tween deck, 9 ft. 6 in.; bulkheads, longitudinal and nine transverse steel bulkheads; holds, 5; hatches, three 31 ft. 6 in. by 20 ft., one 29 ft. by 20 ft., one 27 ft. by 20 ft., one 13 ft. 4 in. by 10 ft. 7½ in.; engines, B. and W. single 4-cycl. Diesel; horsepower, brake 1850, ind. 2250; diameter of cylinders (6), 740 m.; length of stroke, 1500 m.; speed, about 11 knots on about 7½ tons oil; winches, eight 3½-ton, three 5-ton, electric; electric auxiliaries; one ordinary vertical donkey boiler, 100 sq. ft. heating surface, 100 lb. pressure.

Publication of this agreement has brought to light that Sir William Petersen, who negotiated the contract with the Canadian Government, had also been seeking financial assistance from the British Government.

It seems that about a year ago Sir William Petersen entered into a contract with Harland & Wolff for the construction of 10 motorships on the Clyde for a sum of about £1,200,000, subject apparently to the successful negotiation of a guarantee of £600,000 from the British Government under the terms of the Trade Facilities Act. Harland & Wolff are reported to have done no work on the boats while the guarantee was hanging fire. Now the Financial Secretary to the Treasury in London announces that the Trades Facilities Committee cannot act upon the original Petersen application because of the new development, and it is unlikely that a guarantee could be recommended upon the facts now disclosed.

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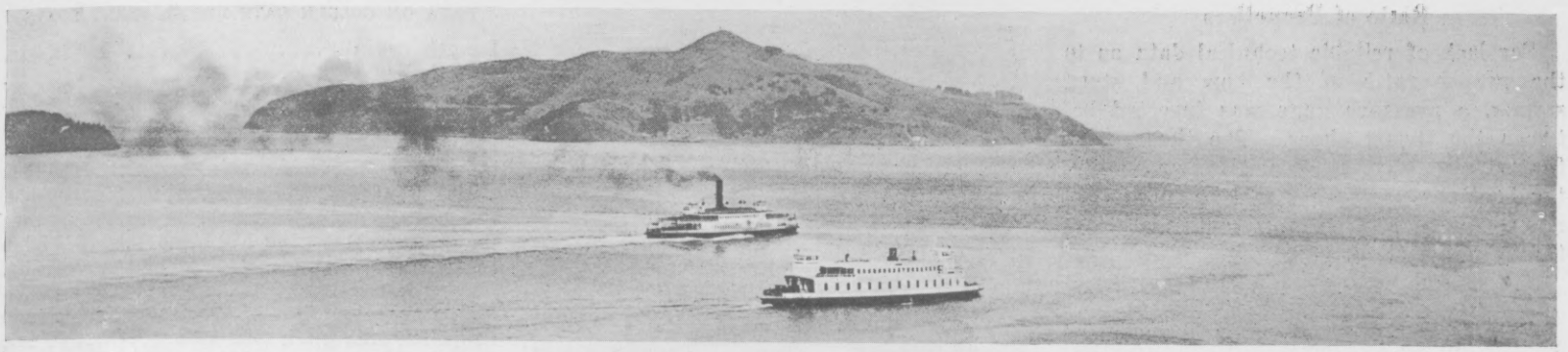

Motor ferry boat and steam ferry boat on San Francisco Bay

Golden Gate Motor Ferry Boats a Proven Success

Report of Operation for the Calendar Year 1924 Tells Exactly How the Owners Feel About These Useful Boats

BY C. C. BROWN*

IN the spring of 1922 the Golden Gate Ferry Company started operations on San Francisco Bay with one steam ferry boat having a capacity of 50 autos. In July of the same year the first Diesel engined ferry boat was put into service by this company. The boat had a capacity for 75 autos and was equipped with two 525 b.hp. Pacific Diesel engines, which operate two 360 kw. G. E. generators. The propelling motors are one of 750 hp. on each end.

Many troubles of a minor nature were encountered, but none of sufficient magnitude as to discourage the directors of this company in placing the contract for a duplicate boat thirty days later.

A 30-minute schedule was maintained by these two boats over a period of 16 hours a day, and in April, 1923 the second Diesel engine boat was put into service.

The following is a summary of operation for the year 1924:

	Total Hr. in Service	*Avg. Hr. Per Day*	*Avg. Fuel Per Hr., Bbl.*	*Total Lub. Per Hr.*	*Total Knots*	*Total Autos.*	*Total Trucks*
Str. Hanford....	2,116	5.8	4.50	$0.09	7,252	72,340	4,914
M. S. Golden Gate	6,115	16.8	1.08	0.41	23,748	193,885	22,143
M. S. Golden West	5,856	16.0	1.12	0.39	22,838	192,931	23,079

Fuel Oils

A successful attempt was made to use heavy fuel in these engines with entirely satisfactory results, the only difficulty experienced being with excessive dirt. Not being equipped with the proper apparatus for cleansing, and in view of the fact that the company was protected with a contract for Diesel fuel and not heavy fuel, the rise in the oil market rendered it unnecessary to go further. It is the intention, however, to install the necessary apparatus soon and no difficulty is anticipated in using a commercial of 14 deg. Beaumé plus.

No attempt has been made to arrive at the fuel consumption per b.hp.-hr. over any long period of time, on account of the variable load when ships are working ahead in the slips. But an occasional check over short periods of time indicate that this is consistent, being obtained with other installations of this type. It is approximately 0.42 lb. per b.hp.-hr.

Lubricating Oil

Satisfactory lubrication is being obtained with a straight-run, pale Western oil of 600 vis. at 100 deg. Fahr. Much criticism has been offered by oil men of the paraffine variety in this respect, but after 18 months of continuous use of this oil the cylinders are still intact and the most that has been taken out of the running parts is 0.003 in. It is interesting to note that prior to the installation of oil coolers this oil entered the engines at 130 deg. Fahr. It now enters at 100 deg. Fahr.

Some difficulty was experienced for approximately one year in getting the centrifuges to function and precipitation was resorted to for lubricating oil clarification. Since that time the manufacturers have put their centrifuges into satisfactory operation. From an oil conservation standpoint the indications are in favor of precipitation, it being understood, however, that no attempt was made to precipitate the carbon in suspension. This at times was as high as 10 per cent by volume.

Maintenance

No difficulty is being experienced in maintaining these boats entirely with the engine room personnel during the daily period of lay-up and the semi-annual docking and cleaning.

Electrical Equipment

Aside from an occasional adjustment of brushes and contacts there has been no expenditure of money or energy on the electrical equipment.

Pilot House Controls

During the year 1924, there were some 25,000 maneuvers made by both boats without a single mishap or replacement to the apparatus.

*Superintending Engineer of Golden Gate Ferry Company, San Francisco.

One of the motor ferry boats of the Golden Gate Ferry Company which are operating so successfully at San Francisco

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Ratio of Propellers

For lack of reliable technical data as to the proper ratio of the bow and stern screws, a pressure gage was inserted between the thrust shoes. The theory was that at a point of no thrust for or against the boat the pressure would be in equilibrium. Constructive comments on this theory are invited.

Governors

Considerable difficulty was experienced for a short time because of the engines racing. This was overcome, however, by the introduction of a solenoid that operates in conjunction with the controls. At present the variation of engine speed during maneuvering is so slight that it cannot be detected in the ships' lights, which are operated from the main engines.

There is no doubt in the minds of the Golden Gate Ferry officials about the efficiency and adaptability of this type of propulsion for double-end ferry boats.

End view of GOLDEN WEST

DATA ON GOLDEN GATE DIESEL FERRY BOATS

Length overall	220 ft. 0 in.
Length b.p.	208 ft. 0 in.
Beam molded	44 ft. 0 in.
Beam overall	60 ft. 6 in.
Draught	10 ft. 6 in.
Displacement	1030 tons
Capacity of fuel bunkers	100 bbl.
Total fresh water carried	4 tons
Brake horsepower each engine	525 b.hp.
Propelling shaft power	750 s.hp.
Engine speed	225 r.p.m.
Ship's speed—actual in service	13.6 knots
Propeller dia., pitch and area	Dia., 96 in.; pitch, 106 in.; area, 5½ sq. ft.
Daily fuel consumption	16 bbl. in 16 hr.
Daily lubr. oil consumption	8 gal.
Number of engine room and boiler staff on watch.	3
Total cost of ship	$137,000
Engine dimensions	15 in. by 25 in.
Cylinder bore	15 in.
Weight of main engines	48 tons each
Weight of generators	13 tons each
Weight of motors	17 tons each
Weight of complete machinery with shafting	156 tons
Length of machinery space	40 ft.

Defects of M. S. Carriso Eliminated by Overhaul

Oceanic Steamship Company's Vessel Used Between San Francisco and Hawaiian Islands Is Now Operating Satisfactorily

"CURING the CARRISO" might well be the title given to this story, which should be read by those inclined to consider early unfavorable reports of motor-converted steamers as valid objections to Diesel propulsion in general. As a result of a number of changes, the CARRISO can now be classed among the non-stop performers.

A few details from the log of its first performance in the year 1925 will indicate the present situation: average speed from Hilo to San Francisco, 9.7 knots; distance, 2004 miles; running time 8 days 13 hr. 12 min.; average fuel consumption of main engines, 52 bbl. a day; stops, none. The entire run from San Francisco via the Columbia river to Honolulu and return was a non-stop affair throughout, except of course for cargo handling.

The CARRISO is a twin-screw motorship with two Pacific-Werkspoor oil engines of 850 hp. each. It was formerly the steamer MAGUNKOOK, built at Long Beach, Cal., in 1919 and converted at the Moore Shipyard, Oakland, Cal., in 1923, for use by the Oceanic Steamship Company as a sugar carrier between the Islands and San Francisco. Considerable engine trouble developed on the first trip to Manila and on subsequent trips to the Hawaiian Islands, while on one occasion a breakdown necessitated her being towed into port.

The Oceanic Steamship Company then called in C. C. Brown, superintending engineer of the Golden Gate Ferry Company, and entrusted to him the general supervision of putting the CARRISO in good shape. It is of interest to note at this time that more money was spent on the steam equipment and cleansing of the engine room in general than was spent on the prime movers.

A general overhauling under the direction of the chief engineer, J. L. Pearson, and the first assistant, A. O. Hinsch, two men of wide experience and thorough understanding of Diesel engine problems, who have handled the CARRISO on its last two trips, resulted in the straightening out of various kinks and the elimination of a number of defects. The steam engines which operated the various auxiliary services, such as the steering gear, sanitary pump, bilge pump, fuel oil transfer pump, ice machinery, electric lighting, etc., were removed and electric drive substituted through a 50 kw. Westinghouse generator and a 75 hp. Pacific Diesel engine. A No. 6 Sharples centrifuge was installed which only had to be run 11 hours a day in order to supply the main engines without heating the fuel oil. Variation of from 0.035 to 0.148 in. diameter in the cylinders was completely remedied by the use of Double Seal piston rings. Despite this wide range, there were no pockets, the taper being gradual. All this work was done by the engine room staff with only minor assistance from other employees of the Oceanic Steamship Company and none from the outside at all.

After the equipment had been put in perfect order, the voyage was made with the results noted above. The record is of particular interest, in view of the fact that instead of regular Diesel oil, heavy boiler fuel was used throughout the voyage, both at sea and while maneuvering. As a matter of fact, on a maneuvering test in San Francisco Bay after the conclusion of the trip, the oil was used cold with good results. At sea a moderate temperature—100 deg. F.—is used, the heating being done by electricity. Another point of interest is that for a considerable period the cylinders were lubricated on red engine oil of a western variety, with entire satisfaction. This was done as an experiment.

Perhaps the most interesting experience of all on the CARRISO was the remarkable success attained with the exhaust valves as a result of the use of a method of blowing by, developed by Mr. Hinsch. One of the valves (of the ordinary cast-iron type) shows no need of attention after 37 days of actual running, whereas on previous trips the valves had to be ground after ten days. All the rest of the valves made the 25-day round trip without being touched, although they were not ground in the first place. On previous trips of the CARRISO the valves had to be taken out and thoroughly overhauled after 13 days.

Before joining the staff of the Oceanic Steamship Company, Mr. Pearson was first assistant on the CHALLAMBA and chief engineer on the ANNIE JOHNSON and the motorship SIERRA, while Mr. Hinsch was first assistant on the ADMIRAL PEARY and chief engineer of the DONNA LANE.

Contract for the two 70-ft. Diesel-electric sternwheel towboats for use on the Kentucky and Ohio rivers has been let by the U. S. Engineer Office, Cincinnati, to the Nashville Bridge Company of Nashville, Tenn., who were the lowest bidders. The main engines will be Winton, and the auxiliary lighting sets will be driven by 12 hp. Hill Diesel motors.

For new construction under the Navy Department the last Congress made appropriation for six oil-engined river gunboats for service in China in addition to submarine work. Various statements have been telegraphed out of Washington regarding decisions to have all the oil-engines built at a government yard. The Navy Department has not yet replied to our question for information on the subject, and we cannot therefore, confirm or deny any of the published statements.

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Striving for 80 Per Cent Propeller Efficiency

Development of Kirsten-Boeing Propeller May Yield a Propulsive Efficiency Heretofore Considered Unattainable

A NEW device for applying power to propulsion of vessels has been invented and developed by F. K. Kirsten, Professor of Electrical Engineering at the University of Washington, working in conjunction with W. E. Boeing of Seattle. This invention, called the Kirsten-Boeing propeller, has undergone extensive tests on a 38-foot motor boat.

It operates on radically different principles from the conventional screw propeller. Perhaps the best idea of the actual construction and appearance of the K-B propeller can be obtained from the diagram and illustration reproduced with this article. The mechanism is comparatively simple. It consists of a large disc installed on the bottom of the boat and driven by spiral bevel gears from the engine. Extending downwards through this disc are several blades which not only revolve with the disc, or rotor, but themselves are geared so as to rotate independently, in the manner of a sun and planet motion.

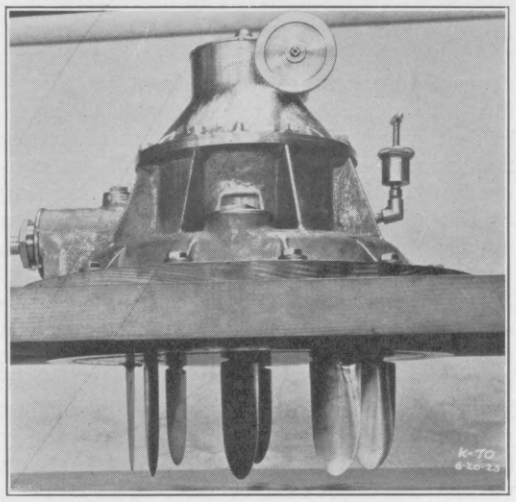
The thrust upon the water is produced by these blades through the combined motions. First, the axes of the blades are caused to move in a circular path with a uniform motion, the plane of the circular path being at right angles to the axes of the blades. Second, while moving in the circular path the blades are caused to rotate on their own axes, this motion also being uniform. The motions are so related that during the time each blade, starting from a given point, makes one revolution around the center of disc, it must make one half a rotation on its own axis.

A report issued by the builders of the K-B propeller explains their operation as follows:

"A condition resulting from these motions (the two motions of the blades on the rotor) is that the plane of each blade (i.e., a plane which passes through the axis and edges of the blade) will pass through one point, this point being located in the circular path on which the blades move and which, when set by the controls, is stationary with respect to the motion of the blades.

"A second condition which results from the two motions of the blades is that, if a line be drawn through the point on the circular path and also through a point diametrically opposite on the path, the corresponding instantaneous positions of each blade on opposite sides of this line will be symmetrical with respect to it. This line is caller the axis of symmetry, and the thrust produced by the blades while being moved as described is always at right angles to this axis and parallel to the plane of the circular path, the direction of the thrust with respect to the sides of the axis being determined by the direction of rotation of the blades around the center of the circular path.

"The axis of symmetry can be turned about the center of the circular path, the center acting as a pivot; hence the direction of the thrust can be turned to any direction desired. It is apparent on following the motions of the blades that the axis of each blade crosses the axis of symmetry twice in one revolution and, at the point on the circle where all of the planes of the blades meet, each blade, at the instant it passes this point, will become tangent to the circle. At the second point



Kirsten-Boeing Propeller Unit

the plane of the blade coincides with the axis of symmetry."

It is apparent that with such a mechanism, no rudder or any of the customary steering apparatus will be required, because the boat can be steered by shifting the direction of the thrust. Similarly, no reverse gear is needed, because the boat can be driven astern by throwing the thrust forward. In other words, propelling and steering are taken care of by one mechanism, simplifying control of the vessel.

The advantages claimed for the K-B propeller grow mainly out of the increased maneuverability it gives the boat, a saving in operating expense due to greater efficiency and a simplification of hull construction.

Since with the K-B propeller the steering of the vessel can be accomplished by changing the direction of thrust of the propeller itself, the pilot has remarkable control over the boat. For any size craft the control is entirely in the hands of one man. The only operation required on the engine is to increase or decrease the revolutions as the engine runs in the same direction at all times.

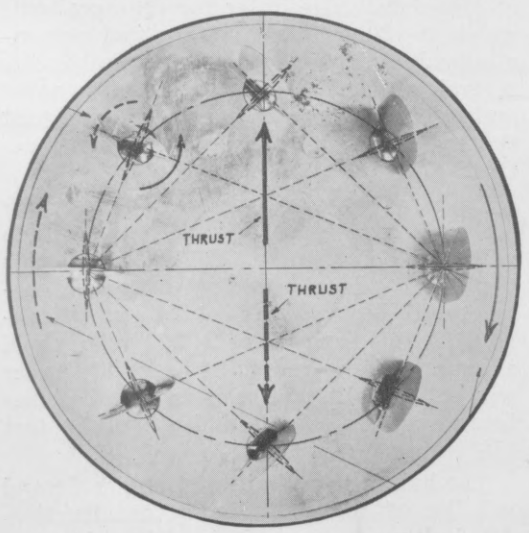


Diagram of Blade Thrust

Full line arrows relate to one direction of motion; dotted arrows relate to the opposite direction

With the high speed test boat it was found possible to turn in a circle of 100 feet radius while under full power. In making a turn the boat banks inward instead of outward as with a screw propelled boat. The boat can be stopped in two lengths and started full speed astern.

The K-B propeller is feasible for twin installations, according to its inventor. Either two engines may be used or two wheels may be geared to the same engine. With a twin installation, the maneuvering advantages of the propellers are very marked. There is practically no vibration and the propellers have a marked stabilizing effect on the hull.

The basis upon which the inventor claims reduced operating costs in the use of the K-B propeller are derived from test runs made with the 38-foot motor boat in which the propeller was first used. The boat is equipped with a 200 hp. marine engine developing its rated horsepower at 1800 r.p.m. The motor is connected to the propeller by means of a shaft through two flexible couplings, one at the motor and one at the propeller. A clutch is provided on the motor to disengage the motor from the propeller, but the reversing mechanism is not used.

The present propeller has been in service since April 7, 1923, and has propelled the boat a distance of 2500 miles, in both fresh and salt water. The propeller so far has not given the slightest trouble and has never failed to complete any trip required of it. On a number of its trips the boat has come into contact with floating logs and boards and bent the blades, but the propeller itself was never damaged. With this machine the operation of taking out a bent blade and inserting a new one is only a matter of a few minutes' time. It is done while the boat is in the water, it being unnecessary to drydock the vessel.

In addition to the mileage made, the propeller has been subjected to many other severe conditions, such as the sudden reversal of thrust and the quick application of full power. After being taken down on having run the first 1200 miles, the parts were found scarcely worn at all and all the gears were in good condition.

On several tests made on Lake Washington, Seattle, over a measured course, the slip of the propeller was found to be 20.1 per cent which, on the basis of tank experiments was deduced to be equivalent to a propeller efficiency of around 80 per cent, a gain of about 15 per cent over the efficiency of the screw.

Having no shaft projecting outside the hull and no propeller blades rotating in a vertical plane, it is not necessary with the

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K-B installation abruptly to draw in the lines of the stern. They can in general be made to run out almost straight, thus simplifying the construction and increasing the space within the hull.

Efficiency tests of a model K-B propeller were made at the Navy Yard, Washington, D. C., under the direct supervision of Rear-Admiral D. W. Taylor, retired. Because of the time limit placed upon the use of the model testing basin the tests originally contemplated could not be carried out so that, unfortunately, no data can be given from tests made with the propeller placed in a model hull.

According to Admiral Taylor's report the diameter of K-B propellers would be rather large to obtain a greater efficiency than the screw for slow-speed vessels, such as cargo vessels, but to quote from the latter part of his report: "For vessels of the wide flat stern type and fairly high speed it seems possible to fit K-B propellers with efficiency markedly superior to that of the screw propeller."

One of the next steps of the Kirsten-Boering Engineering Company is the building of a seventy-five-foot boat exactly similar to the type the Government is having made for Coastguard rum chasers. This vessel will be equipped with the same engines used by the Coastguard, but in it will be installed twin Kirsten-Boeing propellers. The craft will then be used to demonstrate the efficiency of the new propellers.

It will demonstrate, according to Professor Kirsten, that such a craft can be operated for one-half the fuel cost of driving screw propellers, that the new type propeller is a rudder as well as a propulsion mechanism, that its upkeep is virtually nil, and that the vessel can be maneuvered with the ease and facility of an automobile.

This style of propeller is to be installed and tested on the Navy dirigible SHENANDOAH, according to Seattle reports.

Westinghouse Offers Valuable Book

What forms the most comprehensive work on all branches of electrical marine engineering yet brought to our attention is a new and excellently illustrated 528-page book issued by the Westinghouse Electric & Manufacturing Company. The chapter devoted to Diesel-electric drive is one of the most complete sections of the book. Not only are the functions and design of the various parts of the drive mechanism described, but the merits and economics of the system are carefully gone into. Even the motorship operating engineer has not been overlooked, and there is a section covering questions and answers on running oil engines. Space limits prevent our referring in detail to all the useful subjects in the book. Those of our readers desirous of receiving a copy of this valuable volume should make early application to Peter Sensenig, Westinghouse Electric & Mfg. Co., Room 1623, 150 Broadway, New York, N. Y.

After receiving the bids for the conversion of the Shipping Board vessels COURAGEOUS and TRIUMPH to Diesel power, the Matson Navigation Co. of San Francisco decided not to proceed with the project.

Two Inland Motor Ferries Ordered

THE newly named F. M. COOTS, a side-wheel ferryboat owned by the Falls Cities Ferry and Transportation Company of Jeffersonville, Ind, is to have the first application of Diesel-electric propulsion to a ferryboat of this type in America. This boat, F. M. COOTS, formerly the W. S. MCCHESNEY, JR., has been in passenger and freight service between Louisville, Ky., and Jeffersonville, Ind., on the Ohio River since 1912, using reciprocating engines for motive power. The new equipment will consist of two Fairbanks, Morse Diesel engines, direct connected to two 175-kilowatt, 230-volt, direct current generators. Power from the generators will be used in driving two 180-hp. motors, running at 425 r.p.m. at full speed. Through a system of double reduction gears the motors will be connected to the two paddlewheels, turning at 14 r.p.m. The main generators will also supply power for the illumination of the boat and for running the boat's auxiliaries. All the electric equipment will be of General Electric manufacture. Economy is the outstanding advantage claimed for the new type of propulsion. By operating with less fuel, more trips can be made without refueling and there should be no standby losses such as are experienced by steam vessels when docking or idle. The electric drive is also expected to afford greater flexibility in maneuvering and a higher average ship's speed. Absence of cinders and dust is also an advantage in the use of Diesel-electric propulsion equipment. A single generator only will be needed for running the boat at low speeds. Further economies are expected from this. By the use of but one generator, approximately 75 per cent of full speed can be attained, the current from the one machine being used to operate both propelling motors. The boat, with its new electric drive, is expected to go into service by May 1, 1925.

Ferry service between Brockville, Canada and Morristown, carries a heavy tourist traffic during the summer months, and the Brockville & Morristown Co. are adding a new steel motor ferry to their equipment for the coming season. Eads Johnson of New York, has designed a double-ended ferry boat, 100 ft. overall, 40 ft. beam and 5 ft. 6 in. draft, to accommodate 25 cars and 250 passengers. Mr. Johnson, who has had considerable experience in the design of ferry boats of this class, has developed a very serviceable and practical vessel, which is to be constructed at the T. A. Kyle Shipyard, City Island, New York, and is expected to be ready for service toward the end of June. The propelling machinery consists of a Nelseco direct reversible engine of 180 hp. which will drive both the forward and stern propellers simultaneously, reversing to be accomplished by reversing the rotation of the engine. After completion of vessel in New York, she will be run via the Hudson River and New York State Barge Canal to the St. Lawrence River and immediately engage in the regular 24-hour service. With this modern equipment the Brockville & Morristown Co. will be in a position to handle increased tourist traffic without delay.

600 S.hp. Diesel Tug for Honolulu

THE largest twin-screw Diesel towboat ever built on the Pacific Coast for overseas interests and the first ocean-going tug to be built in Seattle in twelve years is now being constructed at the yards of the Ballard Marine Railway for Young Bros., Ltd., of Honolulu. The order for the new boat, which is 120 feet long, was placed personally by Jack Young, who was in Seattle for several weeks. L. H. Coolidge, Seattle naval architect and marine engineer, designed the craft and will supervise her construction.

To be powered with two of the new 300 h.p. Fairbanks-Morse Diesel engines which will give her a speed of 11¾ knots, this tug will handle the towing company's contract with Libby, McNeill & Libby at Honolulu. A fuel capacity of 24,000 gallons will allow a cruising radius of over 7500 miles without refueling. She is 24 feet 7 inches beam and has a molded depth of 14 feet 1½ inches.

The hull is to be built of Douglas fir and will be of unusually heavy construction. All deck machinery will be driven by electricity, current for which will be supplied by a generator operated by a 45 h.p. surface-ignition oil engine. A 7½ h.p. oil engine of the same kind will run the electric light generator. Quarters for 12 men will be provided. The total cost of the craft is to be around $130,000.

Young Bros., Ltd., are engaged in general launch work, towing and freighting in the Hawaiian Islands. The firm was established twenty-three years ago by Mr. Young and two brothers, who later left the organization. The company meets all incoming steamships and takes off the mails, thereby expediting their delivery in Honolulu. It handles the pilots to and from the vessels. In addition to these services, it has the contract for army transportation in the Islands. It is the only towing company in the Hawaiians.

Mr. Young was the first man under the American flag to equip a towboat with a gasoline engine of more than 110 horsepower. The vessel was the tug MAKAALA in which he installed a 140 h.p. Western engine in 1914. The success of this installation was rapidly followed by other large installations, until the company entered the oil engine field. Now the fleet is being converted to Diesel power.

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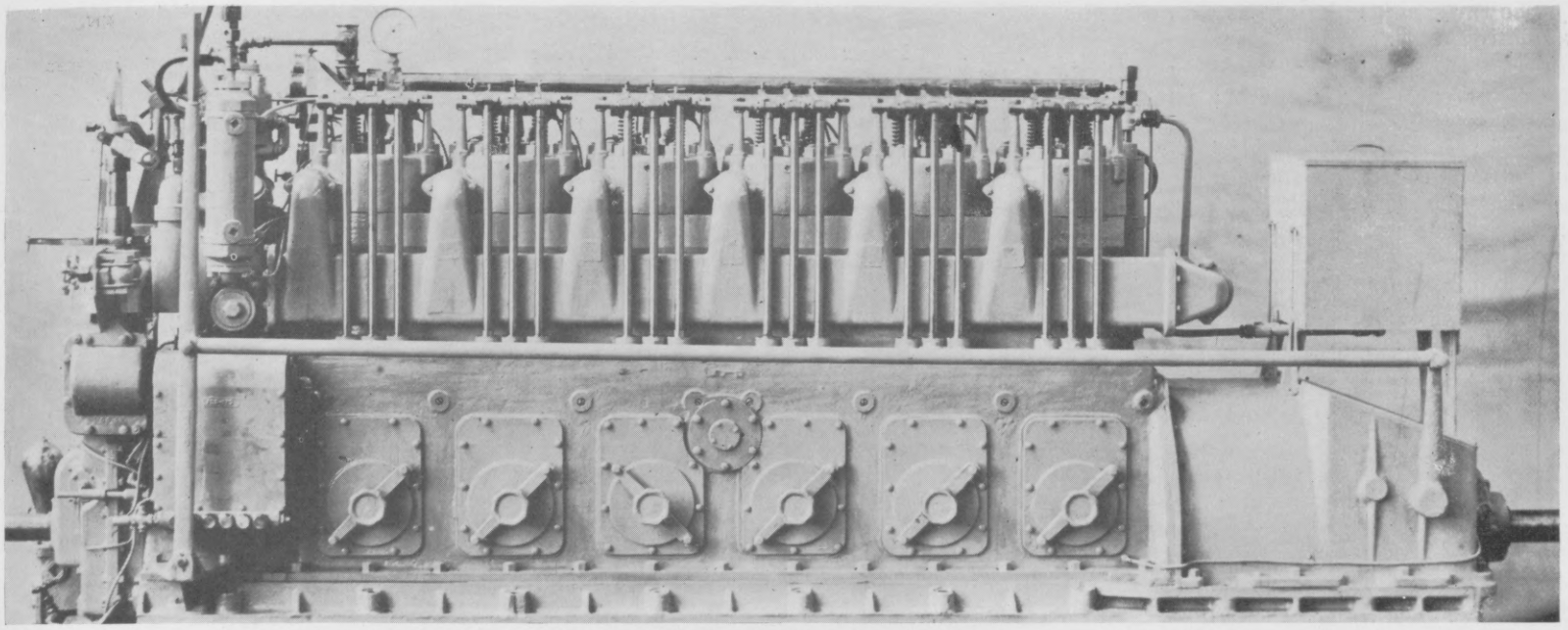


High speed model developing 100 b.h.p. at 600 r.p.m. and utilizing reverse gear system

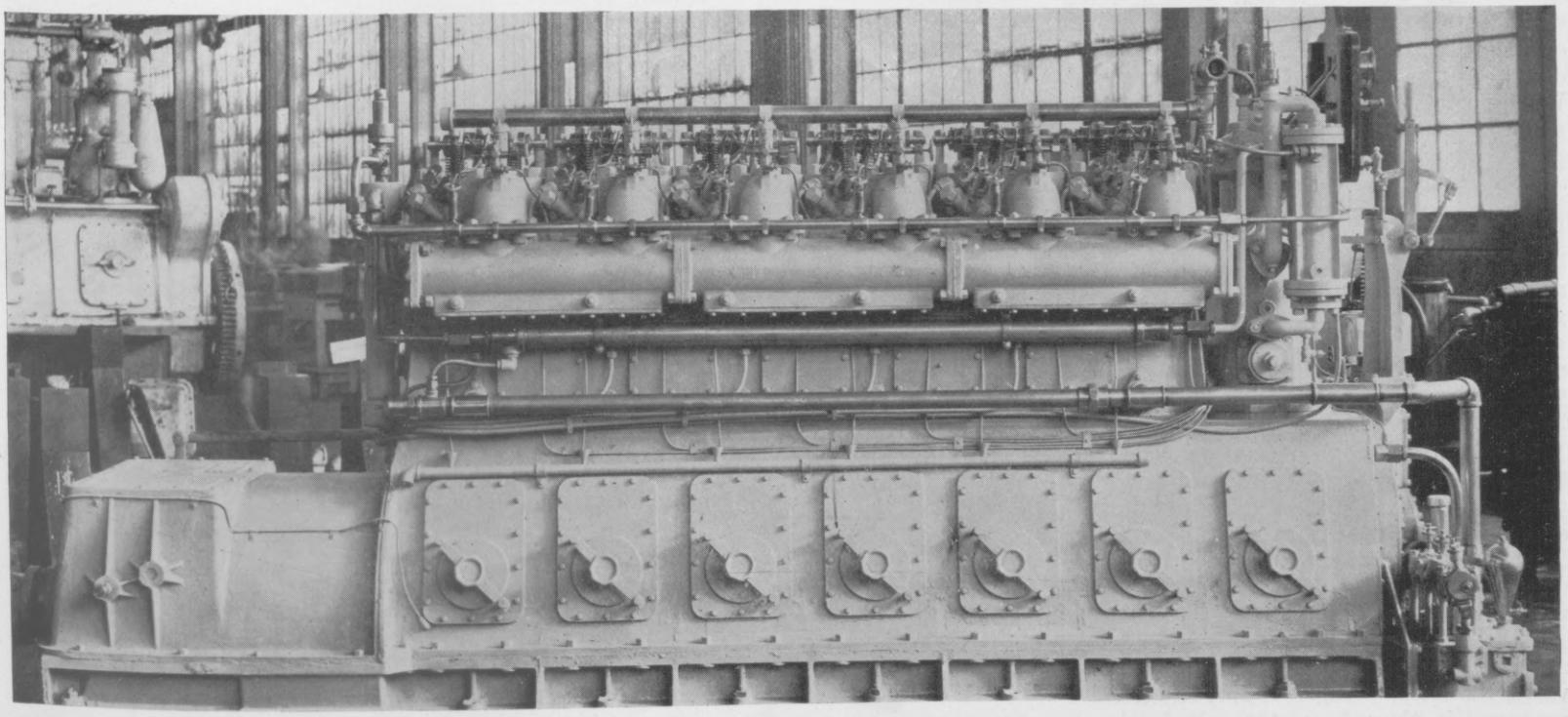
Three New Winton Diesel Marine Engines

Cylinder Block Construction, Air Intake Manifold and Bedded Crankshaft Characterize Latest Design

ENGINEERING is principally responsible for present day progress of the world, particularly in its application to transportation, and as the world advances so must engineering continue to make rapid strides. This factor accounts for the new designs regularly produced by automobile manufacturers, shipbuilders, locomotive makers, oil engine constructors and others, even though their older models may be giving quite reliable and worthy service. Experience provides engineers with countless suggestions for modifications—both major and minor—which entail savings in manufacturing costs at the same time as they result in producing a better product.

In no branch of engineering has such quick development been made as with the Diesel engine, possibly excepting radio which, however, is still in its infancy, whereas the oil engine is now in a substantially commercial stage. The rapidity of its progress has brought many changes in the designs of the leading builders during the past 15 years—especially since the let-up of restrictions that followed the few years since the war's termination.

Three new Diesel models have just been introduced to the marine market by the Winton Engine Works of Cleveland, in accordance with the ever advancing policy set by this company, which appreciates that standing still and resting on laurels is really going astern. One of the new designs was produced to meet the machinery specifications of the 10 Coastguard rum-chasing cutters recently ordered. But it was found that there is a ready market for such an engine and additional orders have been secured from commercial circles for workboats and yachts.



New 150 b.hp. model which offers alternative of direct reversing or gear reversing

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The three new units are as shown in the adjoining table.

One of these engines, it will be noticed, is direct reversible, while the two higher speed sets are equipped with mechanical reverse gears of Winton design and construction. In general these engines follow similar lines of design, although certain modifications are naturally embodied to meet the different conditions of installation and operation.

Model 105—a little, light-weight high-speed unit is of unusual interest, it being the first of its type to be constructed in a commercial way in this country. It develops its rated load of 100 b.hp. at 600 r.p.m. from six cylinders on a weight of 77 lb. per shaft horsepower. While in the main it follows the general characteristics of the bigger engines, there is one important difference, namely, the cylinder block and the upper half of the crankcase with the compressor frame are cast integral and mounted together with the reverse gear on a common bedplate. It might be thought that with an engine of such small size air injection would be dispensed with, but not so in this case, as the Winton Co. is not yet satisfied that airless injection is as satisfactory in an all-round way. While absence of compressor may make a cheaper engine to build, construction costs, however, have been kept down by a simplified arrangement of combined intercoolers and oil separators mounted vertically alongside the three-stage progressive-type air-compressor at the forward end of the engine.

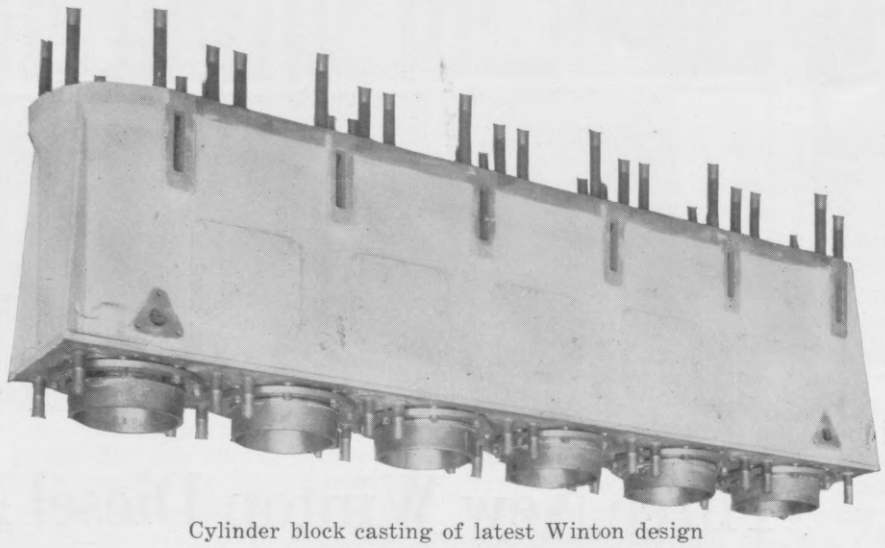
Successful development of this high-speed multi-cylinder engine brings the Diesel engine nearer than ever to the smaller type of pleasure craft such as cabin cruisers, aside from offering valuable experience toward the production of engines for railcars, locomotives, tractors, busses and shovels, all of which are only a stage away from the unit under discussion.

With the two bigger engines the upper part of the crankcase, which in reality is a form of A-frame, is cast separately from the cylinder block. As will be seen from our illustrations the latter follows a successful practice adopted by many of the oldest European Diesel builders such as Werkspoor, Tosi, Burmeister & Wain, etc., and retained in the manufactures of their American licensees. It consists of a tank-shaped casting into which the cylinder liners are inserted. In addition to holding the cooling water this box, running the length of the engine, acts as a beam and stiffens the engine to a considerable degree. The upper part of the liner has a metal-to-metal seating, but at the lower end there is a stuffing-box with regular packing to insure that water will not leak into the crankcase.

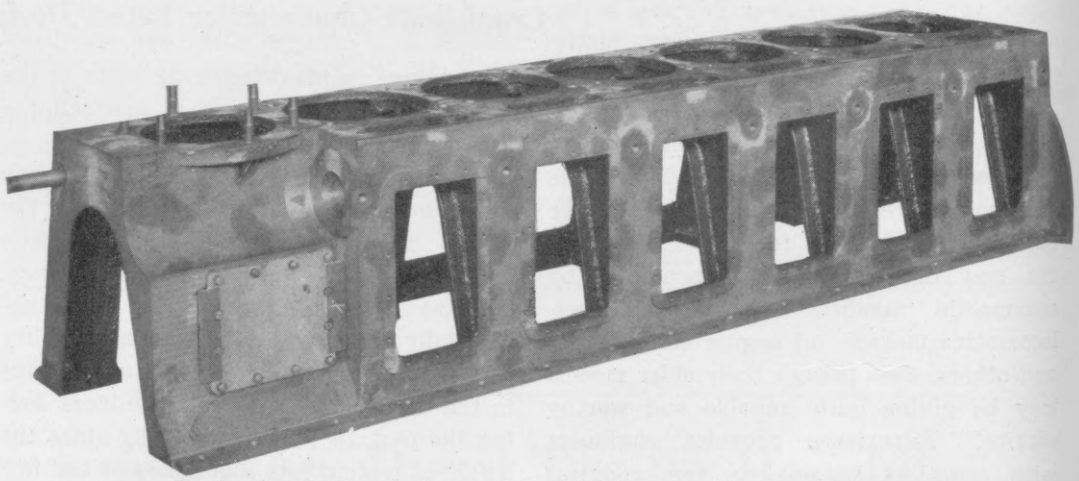
Valve operation, the engines being of the four-cycle type, is by rockers and long push-rods actuated from a camshaft mounted inside the upper part of the crankcase frame, this shaft being driven by gearing off the crankshaft. There are only three rods, for the inlet and exhaust valves and for the fuel valve respectively. There is no rocker or push-rod for the air-starting valve, the latter being a simple check valve arranged in the side of the cylinder head.

SUMMARIZED DATA OF NEW WINTON MARINE DIESEL ENGINES

Model S.C.	Power (s.hp.)	Speed (r.p.m.)	Bore and Stroke (Inches)	No. of Cylds.	Type	System of Reverse	Weight (Short Tons)
105	100	600	6½ x 8	6	4-cycle	Gear	3¾
114	150	450	8 x 11	6	4-cycle	Gear and Direct	6¾
111	200	350	9½ x 14	6	4-cycle	Direct	10½

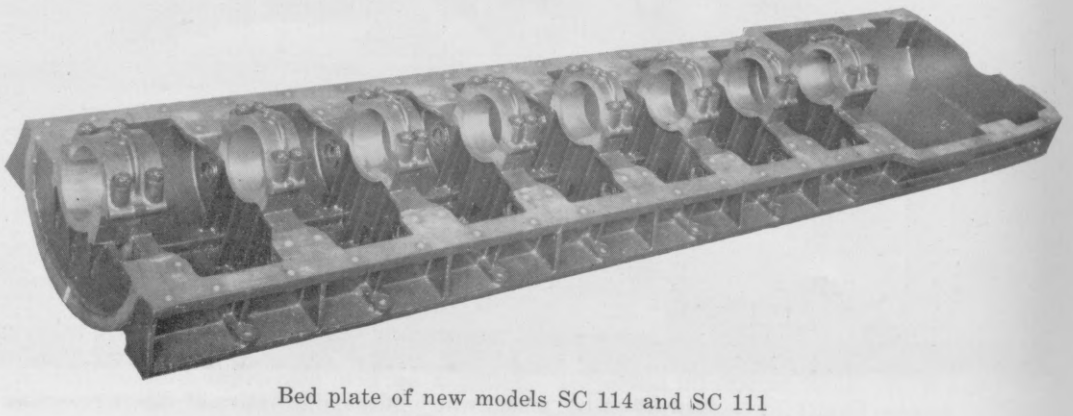


Cylinder block casting of latest Winton design



Single piece engine frame used in 150 b.hp. and 200 b.hp. engines

Crankshaft of these new models



Bed plate of new models SC 114 and SC 111

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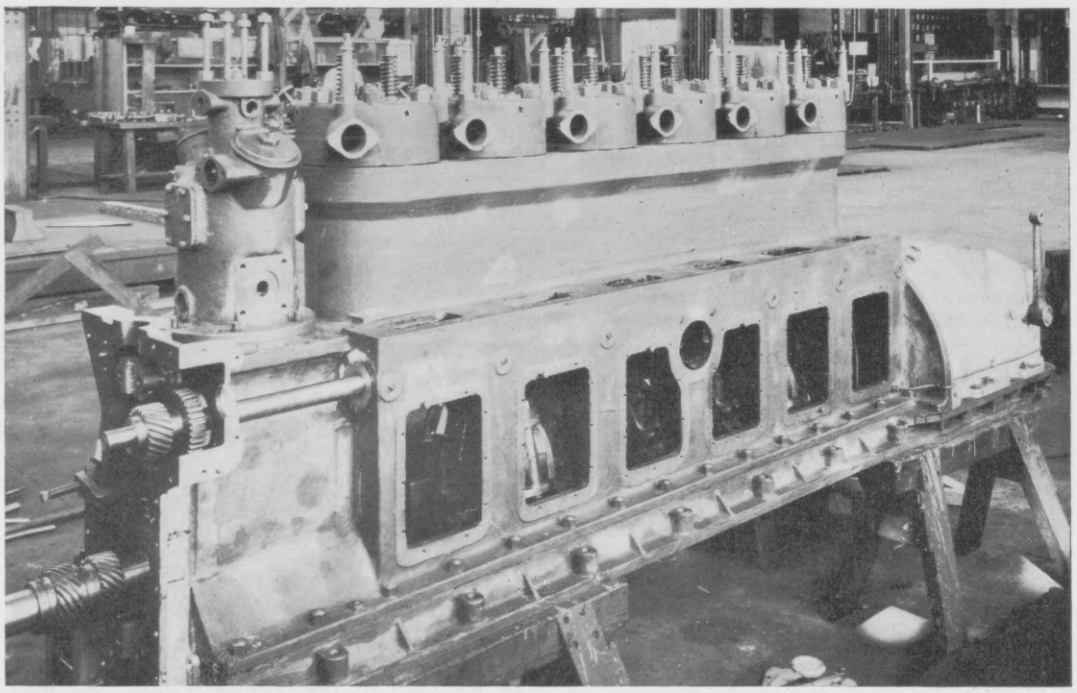
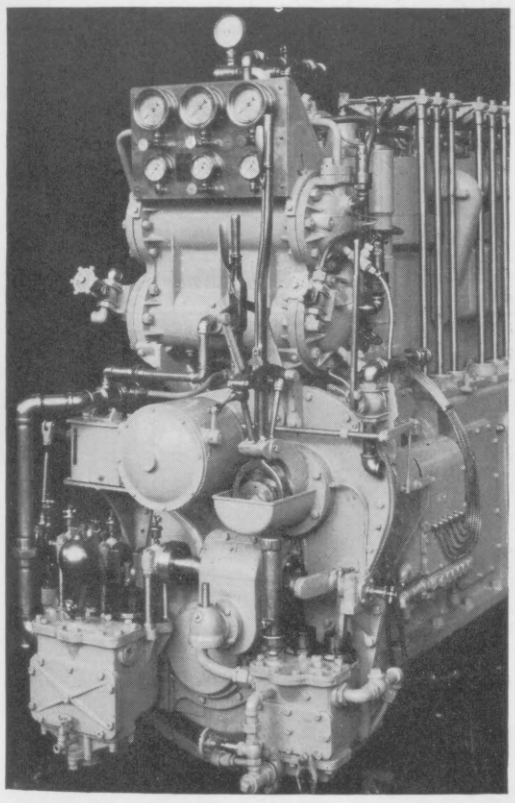
Air is fed to the cylinders when starting, and when reversing in the case of the direct reversible Model 111, by means of a cam operated distributor at the forward end of the engine and which can be seen just over the fuel pump on the inlet side.

A new departure for the Winton engine is the fitting of an aluminum air manifold drawing its air from a common silencer at the after end. This manifold being of flattened shape gives a very neat appearance to the engine. In fact, the engine is unusually clean-looking in its entirety, this being effected by carrying all controls, governor, fuel, water, oil and bilge pumps, etc., to the forward end. It will be noticed in the forward end view of the two larger engines that the arrangement of intercoolers differs from those on the small engine, being one above the other horizontally instead of vertically on either side of the compressor. Reversing of model 111 is carried out by moving the camshaft fore and aft, bringing a second set of cams into operation under the push-rod rollers, the lever operating the same being interlocked with the air-starting lever. The fuel pump has six separate plungers metered by cut-off valves which work on the suction strokes of the pump plungers, being regulated in turn by a centrifugal type governor.



Cylinder block and engine frame cast in one piece for 100 b.hp. engine



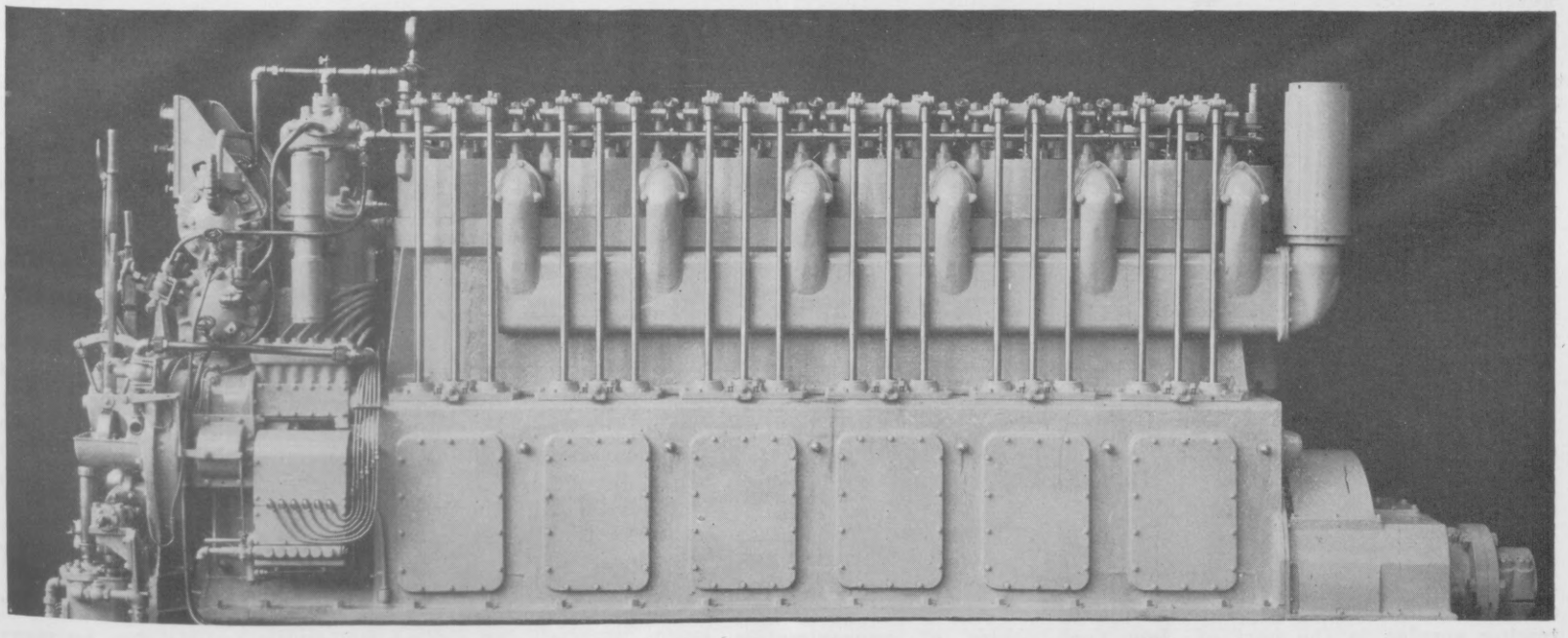
Control end of 200 b.hp. engine

The injection valves are of the mushroom opening downward type which, while adopted for about fifteen years by one of the largest European Diesel engine manufacturers, is rarely to be found in a moderately powered high-speed oil engine—the latter generally being equipped with needle type valves opening upward.

Lubrication is of the force-feed system throughout, the reciprocating pump delivering oil from the supply tank to the header in the crankcase. From this header it is forced through the main bearings, then through passages drilled in the crankshaft to the connecting rod bearings, thence up each connecting rod through a tube to the wrist pin sleeves. Then the oil passes through the wrist pin bearings in the pistons, whence it drains to the sump. The second pump draws the oil from the sump to the oil cooler and finally into a filter consisting of a series of wire gauze screens.

With early Winton engines underslung crankshafts were adopted, but with the new engines, as with other recent models, the bedded form of crankshaft and bearings is now a permanent feature. Crankshaft is carried in eight main bearings cast integrally with the case, but the bearing shells are movable and are of iron lined with babbitt. They are held in place by cast iron caps and steel studs. The bed plate itself is a semi-steel casting, the crankcase and cylinder box also being of semi-steel construction.

All three models we recently inspected when at the Winton plant, where we found an unusually large amount of work on hand. In fact, we are advised that the Winton company already has enough work to keep its shops busy for the balance of the year. While this is largely marine work, quite a number of land installations are under construction, including another set of engines for the Humble pipeline.



Latest design of 200 b.hp. Winton directly reversible marine engine turning at 350 r.p.m.

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Thos. Orchard Lisle.....Editor
Russell Palmer.....Manager

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NEW YORK.....27 Pearl Street
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Cable address—Motorship, New York
Telephone Bowling Green 3420
SAN FRANCISCO.....417 Montgomery Street
Telephone Douglas 6974
SEATTLE.....71 Columbia Street
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Readers are invited by the Editor to submit articles, photographs or drawings relating to motorships, marine oil-engines or auxiliaries. Contributions used in the magazine are paid for on the 15th of the title month of issue, and other contributions are returned as promptly as possible.

Examining Propulsion Systems

There are almost as many varieties of connection between engine and propeller offered the shipowner or boat owner today as there are different types of oil engines from which he has to select. In the course of the last few years we have seen motor vessels put into service with direct connected shafting, with electric drive, geared shafting, hydraulic drive and various combinations of these major classes. The vessels are operating in widely separated parts of the world in different services, under the flags of different nations, with so many other factors of isolation, that no specific knowledge of the relative merits of these different systems has become available. We have learned in a general manner that the specific features claimed for each different style of propulsion have in the main been proved. It is obvious that there are considerable differences in the mechanical efficiencies, but the same variation is shown also in the main engines. This is not really a criterion. These systems, to be compared on a common basis, should work side by side in similar service, under similar conditions and preferably with similar engines. Relative working merits would then be determined by the greater suitability of one or the other of the systems. It looks as if a pretty close comparison of this sort will soon be possible in towboat service in New York harbor. There are already tugs operating with direct-connected propellers and with Diesel-electric drive. A tug with a geared connection will be put into service now that the break-up of the ice has permitted it to be brought down the Hudson. There is the further likelihood that the motor tug fleet in New York harbor will later number among its members a towboat with hydraulic drive. The particular, and in some cases the peculiar, requirements of towing service in the port of New York will enable the merits of these different systems to be tried out and measured according to standards. It may well prove that one system will show up better for one class of work and another system better for another class of work. It is doubtful that one system will prove superior for all classes of work. Such practical experience as may be gleaned from experience in New York will not necessarily apply in other ports and under other conditions. It looks, however, as if for the first time there will be a direct line on the respective operating characteristics of the different systems under one or more specific groups of service conditions.

Instituting Ferry Services

One of the most widely useful services in which the marine oil engine can be employed is the propulsion of ferry boats. A good start has been made. During the last few years, numerous installations have been completed on the Atlantic Coast as well as on the Pacific Coast and also at some points inland. Ferrying makes such intermittent demands upon the power equipment that when steam was the only available power medium, the cost of ferry services was often estimated to be prohibitive. At many places therefore a ferry service is lacking because hitherto it has looked like a losing proposition. The oil engine has changed this. In many places where steam ferries were impossible for business reasons, motor ferries can give efficient service at a profit to the operator. This is being demonstrated on a number of inland waterways and lakes, as well as along the coast. With the vast extension of highways, which of itself is an outgrowth of greater highway traffic, the demands for ferry services are becoming more and more numerous throughout the country. In these days when automobiles form so large a percentage of the traffic a ferry service is not dependent only upon local business. Where ferry services offer means of avoiding detours of many miles, drivers are willing to pay well for the cut-off privilege. States have been making lavish expenditures on highways and have not been very parsimonious about the construction of bridges. It is becoming a problem however, to meet further construction charges and in all likelihood highway construction will slow up. With the decrease in bridge construction and no let-up in the increase of highway traffic there must evidently be many new ferry services instituted and these no longer will be a problem because the oil-engine will enable them to be operated profitably.

Valuable Engine Trials

Publication of the second report of the Marine Oil Engine Trials Committee formed under the auspices of the Institution of Mechanical Engineers on the other side, furnishes a convincing demonstration of the value of the work which the Committee is performing. Many scoffed at the proposal, when it was first mooted, that a series of comprehensive standard tests could be carried out on different engines of approximately the same power and in ships of approximately the same size. Drawing its members from various branches of shipping and shipbuilding, the Committee has not only been fortunate in having men of great ability willing to devote the time to the work, but it has had the advantage of the necessary financial support to pay for the conduct of the most complete series of tests that have ever yet been attempted. When the report of the trials of the SYCAMORE was issued there was evidence that the work had been so thoroughly carried out that a very valuable record had been made available. Now that the report of the trials of the DOLIUS has been made public the shipping world at large will begin to realize much more clearly the great value of all this work. As successive reports are issued in future there will gradually be established a series of test figures and performance data that will be of great use in the development of engine design. The reports will enable engineers to judge better in which direction improvements can be made toward greater efficiency. The information is, of course, chiefly of value in design. It will have little bearing upon construction, nor upon operation, nor maintenance. Inasmuch, however, as design is the starting point, it is right there that the efforts for greater efficiency must commence.

Canada May Try Shipping Subsidy

Although Canada still has its government-owned fleet of steamers the prevailing sentiment in that country seems to be that the Government should get out of the shipping business. On the other hand, strong representations are made from the Western Provinces that private enterprise is responsible for such heavy shipping charges on exported commodities that the producers are deprived of a considerable share of the gains they should reap as a reward for their labor. Private shipping interests stoutly maintain that they do not make greater profits than the risks of their business entitles them to make. This conflict of opinions has involved the whole question of Canadian export shipping in a heated political squabble which reverberates from Ottawa to London. The issue is really drawn between Canadian farming interests and British shipping interests, which makes the problem much more difficult than it would be if Dominion interests alone were concerned. Having failed to give satisfaction to the Western farmers by means of its nationally owned merchant marine, the Government of Canada is apparently ready to attempt another solution, in this case by means of a subsidy to a private shipping firm. The terms under which the Government and the private company finally came to an agreement are worth studying because of the importance of the issue which they attempt to solve. The outstanding feature of the draft contract is the reservation by the Government of complete control over the rates. Whether a shipping service can be maintained on a business basis under such control depends so largely upon political factors that nobody would hazard a guess as to the result. It is interesting, however, to see that such an arrangement has been proposed, and if the agreement is confirmed by the Dominion Parliament in Ottawa its operation will be watched with close attention by many outside Canada.

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Notes and News of the Month from Everywhere

Record of New Construction, Launchings, Trials, Ships' Performances and Other Events at Home and Abroad

Four-cycle A. E. G. Diesel engines of 7000 s.hp. per ship will be installed in the four 10,000 tons 14½ knot twin-screw motorships recently ordered by Furness Withy from the Deutsche Werft of Germany and which will be run under the British flag with British crews.

American shipowners order motorships from Great Britain, and British shipowners contract in Germany for new motorships—both on grounds that construction costs in their own country are too high to enable them to compete in the world's commerce on the high seas.

When they take delivery of a couple of motorships now under construction, the Bergen Steamship Co. of Bergen, Norway, will own a total of eight Diesel-engined vessels aggregating 48,000 tons gross register. The company recently declared a dividend of 8 per cent. The company's fleet consists of 50 ships aggregating 72,689 tons gross register, of which the eight motorships form the principal portion.

CARIA ex FULLAGAR), a boat of 398 tons gross, has been sold by the Manx Isles S. S. Co. of Liverpool to the British Columbian Cement Co. of Vancouver, B. C. She is a welded steel vessel.

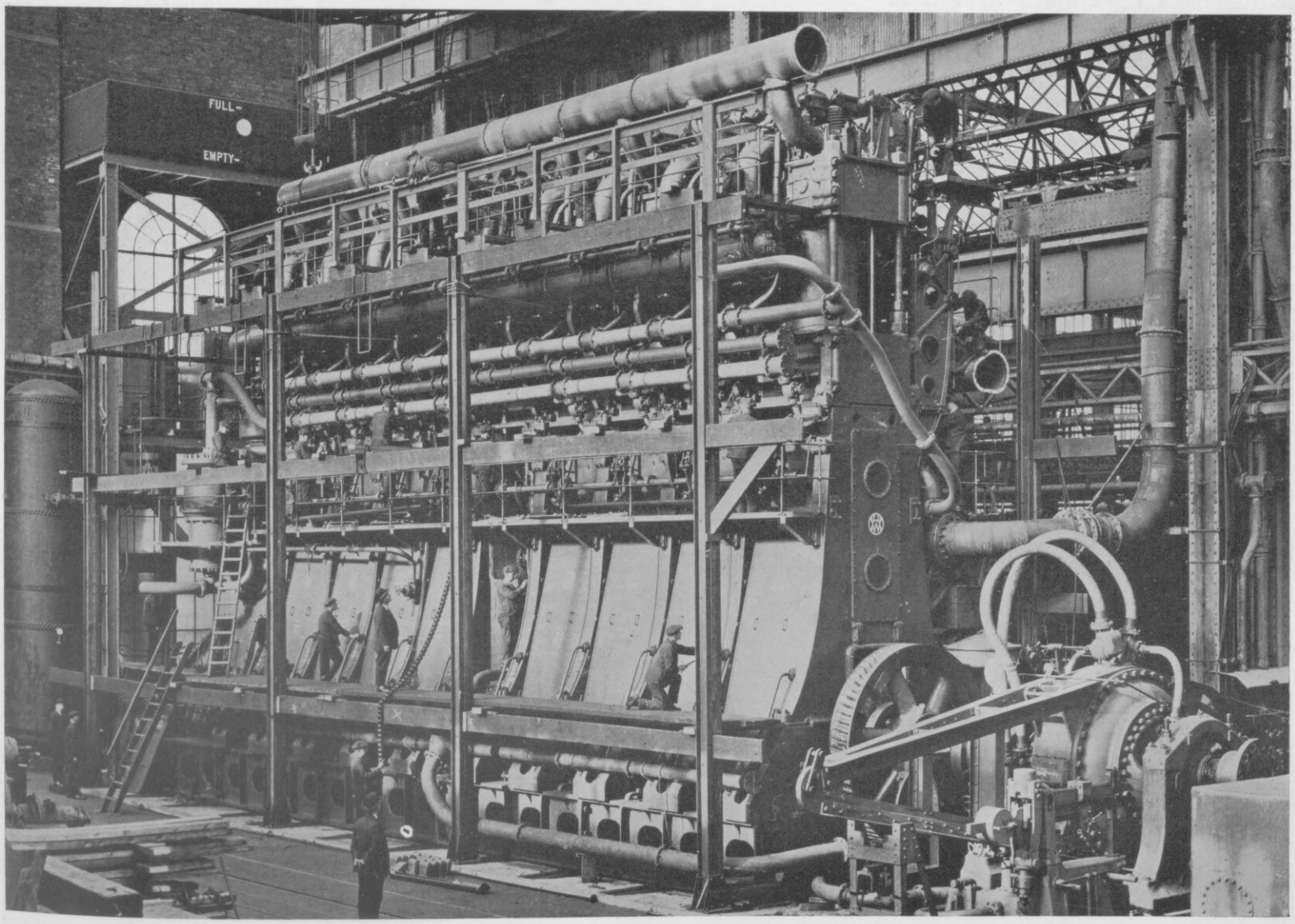
"Very extraordinary absence of vibration in every corner of the ship is a feature of the motor-liner AORANGI," says Robert Traill, engineering director of her builders. "Shipowners could very much alleviate their serious position if they would scrap a lot of their old steamers and give shipbuilders orders for Diesel-engined ships like the AORANGI."

Diesel-electric auxiliaries are being installed on a new passenger steamship building at Swan, Hunter & Wigham Richardson's. So far as our records show this is the first steamer to have her cargo handled by Diesel electric power. An article suggesting that all steamers have the operation of their deck machinery converted to Diesel electric power was published in MOTORSHIP four or five years ago.

According to figures secured by Doxford's on eight motorships which they built, a Diesel-engined ship can transport 100 tons of freight one mile at 10 knots at a cost of one-half cent when fuel costs $8.50 per ton.

Six Rhine service cargo motorships of 723 tons d.w. each are under construction for the Rhenus Transportgesellschaft of Basle at the Gutehoffnungshütte Oberhausen at Walsum. Each is to be propelled by twin 200 b.hp. M. A. N. Diesel engines turning at 250 r.p.m. JUPITER, the first of this fleet craft, is in service and has a speed of 10 land miles per hour.

The tanker MASCONOMO has had her twin-screw triple-expansion steam machinery replaced by A. E. G. Diesel engines to the order of the Julius Schindler Oil Co. of Hamburg. She was built in England in 1904, and was owned by the Standard Transportation Co. of Hongkong, which recently sold her to the Schindler interests.



Harland & Wolff have erected one of the six double-acting engines of about 9000 s.hp. each, building for motor liners

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Sketches and Working of Oil Engines*

Different Types of Engine Framing Avoid "Springing" Due to Engine Loads and Ship's Deflections in Various Ways

ACCORDING to what has been studied thus far concerning the fundamentals of oil engine framework it appears as though the requirement for fastening the frames as closely as possible to the main bearings is of paramount importance. If the comparatively high loads imposed on the structure by the compression and firing pressures are to be prevented from producing troublesome deflections, they must be transmitted in the most direct possible way.

Some engineers, at least, seemed to take this principle so seriously that they thought the framing itself could be eliminated by means of it. They used nothing but steel tie rods similar to those shown in Figs. 38 and 39. There appears to be little doubt that this arrangement effected an extremely direct transmission of the load from the cylinders down to the main bearings. Both the endwise swaying of the engine as well as its tendency to be deflected sideways by the connecting rod thrusts were intended to be taken up by means of the diagonal braces clearly shown in the illustrations.

Effect of Ship's "Working"

There are some observers who claim that the strains which affect the structure of an engine are not confined to those which arise out of the normal working loads and that the deflections of a ship's hull do their share toward wracking the framework. In agreement with this view it has already been pointed out that the performance of engines which had run successfully while bolted down on concrete was different when the same engines were used for driving ships. Reports have been received, also, to the effect that warm bearings occur more frequently in rough than in calm weather.

Some such considerations as these appear to have played a part in the designs of the upper works of many of the engines which are seen in service today. Even in the case of those engines made up of strongly individualized cylinder units an effort has been made to tie the cylinders securely together near the tops, as may be observed by a close examination of Fig. 28. (MOTORSHIP, March, 1925, p. 199.) In designs such as those illustrated in Figs. 35 and 39 an interconnection between the cylinder

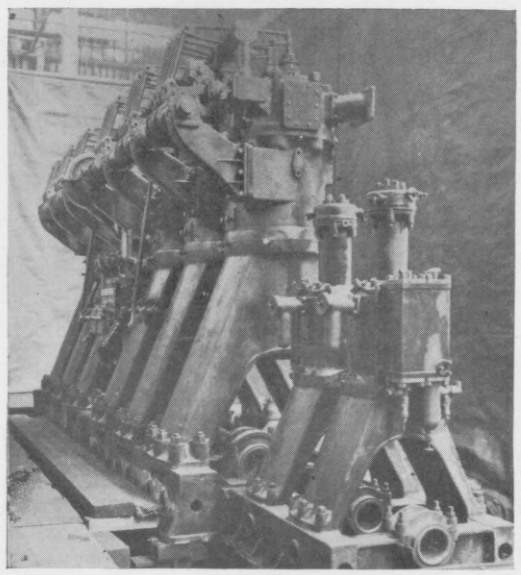

Fig. 40. Individualized engine framing

supports is more or less conditioned by the fact that the lower supports do not come directly under the cylinders. A bridge reaching from frame to frame is necessary, anyway, to form a support for the cylinders and by means of only slight metal additions the bridge is made continuous over the entire engine. Note by way of contrast the engine shown in Fig. 40, which is made up practically of individual pillars mounted on a common bedplate and connected at the top by the camshaft and the exhaust pipe.

Doubt seems to have arisen as to whether unreinforced steel column framing such as that shown in Figs. 38 and 39 would possess a degree of rigidity adequate for marine service, and it is interesting to note that the same manufacturer who formerly built his engines as in Fig. 39 now furnishes engines like those illustrated in Fig. 41.

As has already been pointed out, the use of A-frames generally goes hand in hand with the adoption of a longitudinal cylinder girder like that shown in Fig. 42. It is in some respects the counterpart of the bedplate and adds materially to the strength of the engine. It also functions as a water-jacket.

Cylinders are generally made of simple cylindrical liners fitting into bored recesses in the cylinder beam. Since the endwise cylinder load is transmitted by means of the cylinder head directly to the beam, the cylinder barrel is subjected to bursting stress only. It is held at the joint between it and the cylinder head and is only loosely centered at the lower end, where some form of slidable packing prevents the escape of cooling water.

Whereas the cylinder beam shown in Fig. 42 is cast integral, some manufacturers prefer to make up the beam of individual flanged sections as shown in Figs. 37 and 35 (MOTORSHIP, March, 1925, pp. 201-202). In both cases a deep box section of great inherent rigidity is sought after as a means of making the engine structure as immune as possible to deflections of whatever kind.

Supporting the Crosshead

As has already been pointed out, the type of framing which is interposed between the cylinder beam and the bedplate has an influence on the design of the crosshead which is used; a brief digression on the latter subject may therefore be permissible at this point. If the gib type of crosshead is used in connection with A-frames placed astride the main bearings, some kind of a bridge (Fig. 35, ibid.) must be provided to support it between the

*Summary of a course of instruction at the Polytechnic Institute, Brooklyn, N. Y., by Julius Kuttner, B.Sc., Licensed Chief Engineer, Associate Editor of MOTORSHIP. This is the fourth chapter, the first one having appeared in the January 1925 issue.

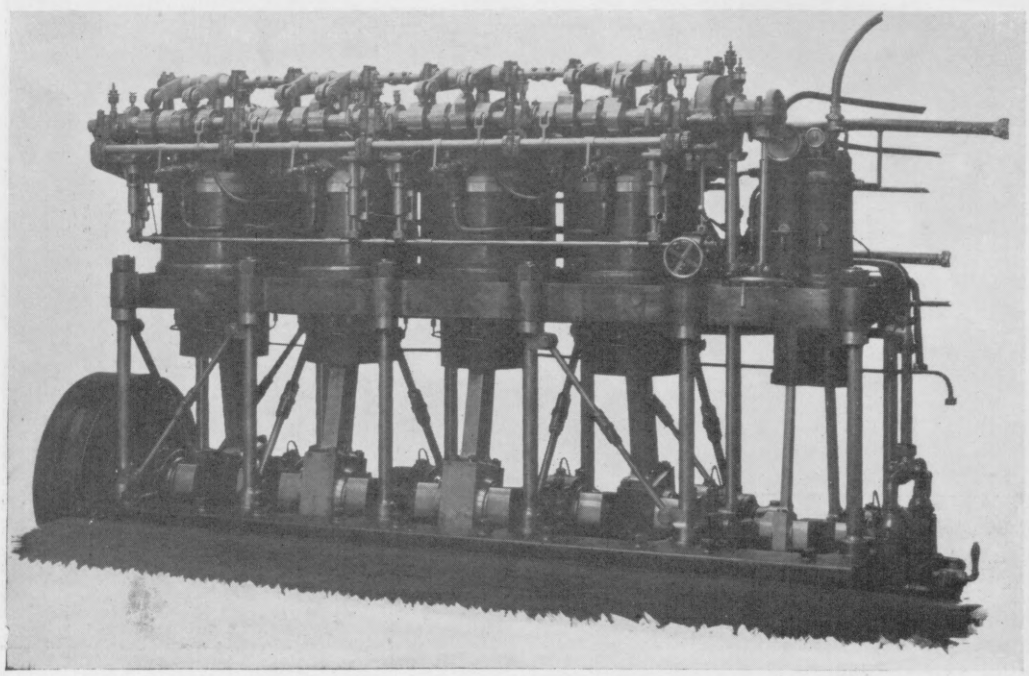

Fig. 38. Framing made up of steel rods taking direct stresses

Fig. 39. Steel tie-rods for crosshead engine

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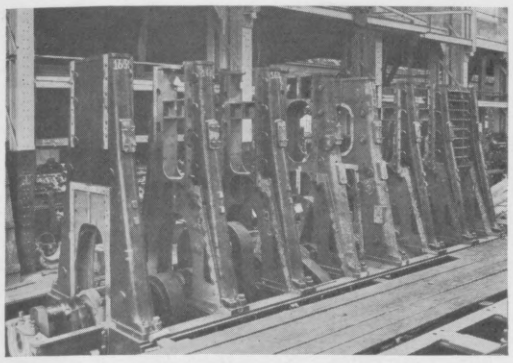

Fig. 41. Two-part A-frames astride main bearings

frames, an arrangement which generally restricts access to the crosshead to only one side of the frame. Space around this part is apt to be limited anyway, particularly for keying up the crosshead bearings.

Partly with a view to increasing accessibility around the crosshead and partly in order to avoid the weight of the bridge between the frames some oil engine builders have revived the older form of steam engine crosshead in which the sliding members are split into two groups and are placed to either side of the connecting-rod fork, Fig. 43. The crosshead pin is here extended far enough beyond the fork bearings to permit of its being centered in each of the half-crossheads, with the result that the pin itself becomes long enough to act as the bridge between the frames. Owing to the fact that the slides are now far removed from the cylinder centerline it becomes possible to abut them directly to the webs of the frames. The latter now take up the crosshead side-thrusts without the intervention of the bridge, leaving front and rear access to the crosshead practically unobstructed (Fig. 44).

Box Construction

Another type of framing with which the double side crosshead is used is shown in Fig. 45. Here again the frames straddle the main bearings, but they are also split along the vertical central plane of the engine for the sake of ease in handling and to permit the crosshead guide flanges to be more easily machined.

It is perhaps not apparent at the first glance that the principles governing the design shown in Fig. 44 and that shown in Fig. 45 are very similar in certain important respects. The resemblance becomes more real when it is considered that the design sketch of the horizontal cross section shown in Fig. 43 applies in principle, at least, to both of them. There can be no doubt that the framing shown in Fig. 45 has the same transverse frame webs as those exhibited by Fig. 44; that the webs are astride the main bearings and that they carry the slides for the two-piece crossheads.

Viewed from a slightly different angle, the framework shown in Fig. 45 may be regarded as an intermediary form between the individual A-frame construction and the box-type of engine housing. The latter may be regarded as having been evolved out of the individual A-frames by the simple process of extending the side-flanges of the latter near their top and bottom ends so far that they meet. Instead of having the wide-open space found between the ordinary type of A-frame (Fig. 44), there now remain merely the oval openings (Fig. 45), such as are more frequently found on the integral box-frames of smaller trunk-piston engines.

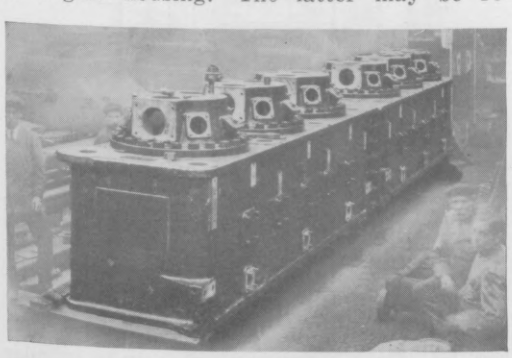

Fig. 42. Cylinder beam for use with A-frames

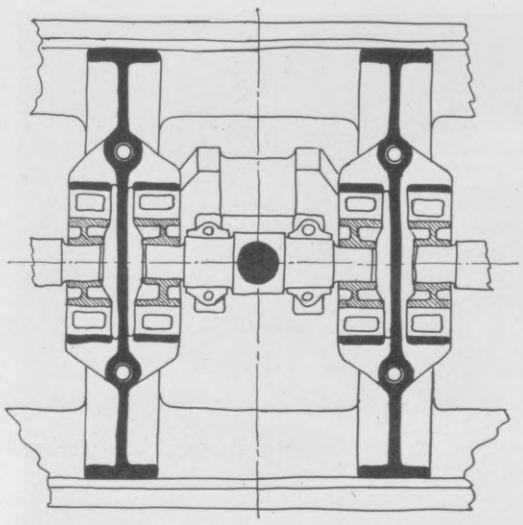

Fig. 43. A-frames with double crosshead

With the joining of the frames in the manner shown in Fig. 45 a more than superficial resemblance to the box-frame has been produced. Note that the frames are also flanged together on the cylinder centerlines and that they make up a rigid unit which is equally independent of the bedplate below or of anything like the cylinder box shown in Fig. 42. Much the same result is achieved by the framing shown in Fig. 46, in which the tops of the frames are joined front and rear by means of stout plates bolted on through the holes clearly visible in the picture. The central idea appears to be that of producing an engine structure of great inherent rigidity capable of maintaining the alinement of the moving parts against the deflections which are encountered in a ship's hull.

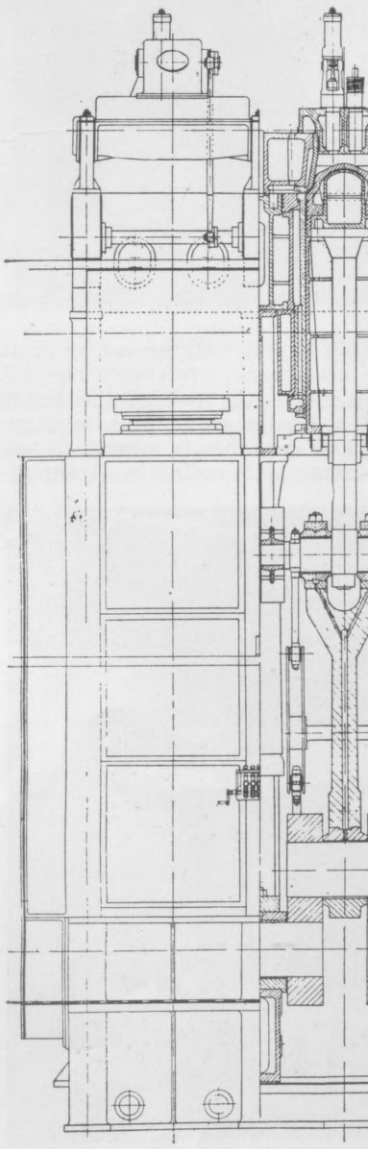

Fig. 44. Side view of double crosshead

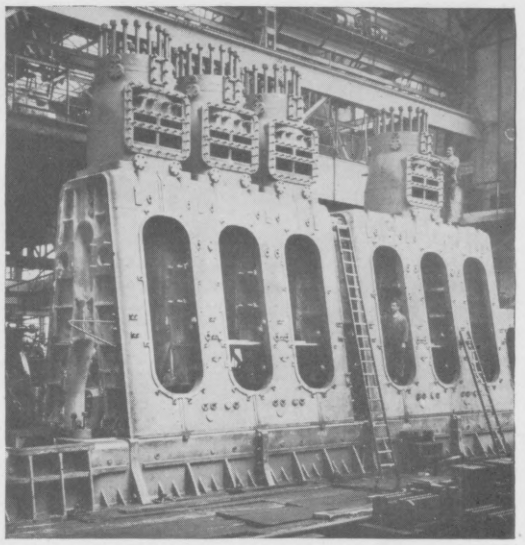

Fig. 45. Box structure evolved out of A-frames

It seems almost superfluous to dwell further on the fundamental resemblance between framing of the type shown in Fig. 45 and that illustrated in Fig. 47. The latter has no crosshead and is cast integral instead of having four matched pieces grouped about the cylinder centerline; but as far as considerations of rigidity and the production of a box-structure are concerned the designs are closely related. Reference to Fig. 48, which shows a bottom view of the frame for the engine in Fig. 47, makes it possible to approach the comparison from the other end. Bearing in mind that the arches which are prominently shown in Fig. 48 between the cylinder openings come directly over the main bearings, it is easy to imagine that the arches are independent castings bolted together in some such manner as that illustrated by Fig. 45.

What appears to be a slight modification of the types just considered is shown in Fig. 49, which may be regarded as an engine like that of Fig. 41 built without a crosshead. It also possesses considerable resemblance to Fig. 47, at least in so far as the crankcase structure is concerned. The greatest difference seems to lie in the fact that the webbed members over the main bearings are bolted to the upper structure by means of machined bolting faces, instead of being parts of an integral casting.

Most of the designs considered probably differ in the location of the joints between the component casting parts of the framework, rather than in the location of the metal sec-

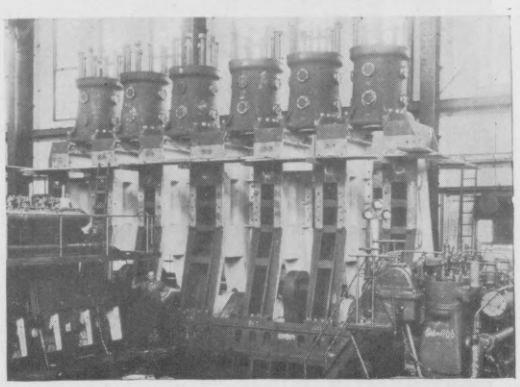

Fig. 46. A-frame tops joined by cylinder castings and plates

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tions. Given a row of cylinder bores and the main bearings belonging to them, there seems to be a quite general consensus of opinion among designers as to the manner in which the framing metal is to be disposed among them. As regards the manner of bolting up, they undoubtedly have their individual preferences.

Tie Rods

It may already have suggested itself to the notice of some readers that tie rods are found on some of the engines used for illustration and are omitted from others. Much could be argued on both sides of the question as to whether tie rods should be used and there are engine builders of established reputation who favor them and who consistently omit them. Theoretically, of course, the steel forgings of which the tie rods are made can resist the tension stresses better than the cast iron, a consideration which appears to have been uppermost in the minds of the engineers who designed the framing shown in Fig. 39.

But it has already been indicated that an engine framework should be capable of resisting loads other than the simple and direct ones. There seems to be need for rigidity to resist the distorting tendency of the engine loads and the deflections caused by the instability of the ship's hull.

Generally speaking, if there is metal enough in the framing to make it adequately rigid, the tensile stress per unit of cross section is considerably below that which can be safely taken up by cast iron. Fastenings at the joint faces must, of course, be provided and the question seems to boil down to a choice between the relatively shallow bolting flanges or the through-running tie-rods. The latter undoubtedly take all the tension stress away from the cast iron, while they have a tendency to produce intense localized strains. Bolted flanges, on the other hand, appear to afford a ready means for distributing the forces more uniformly throughout the castings and for avoiding seriously localized tension. Some tailors sew two pieces of cloth together with a few large stitches of thick thread; others prefer to make a great many stitches with fine thread.

If tie rods are used a special procedure is sometimes employed for setting up the nuts on them. It can well be imagined that screwing home a nut with a 5-in. U. S. standard thread is not an everyday undertaking. Above all, there seems to be need for holding the shank of the rod against turning while the nut is being flogged and for that purpose ordinary feather keys are sometimes provided. If there are no keys, a 4-ft. chain tongs may be applied to some part of the rod which is accessible from the interior of the framing.

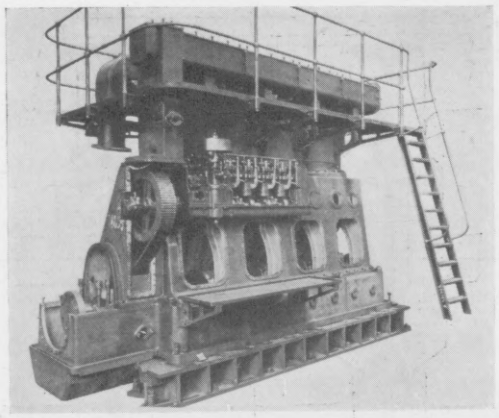

Fig. 47. A-frames in integral box-casting

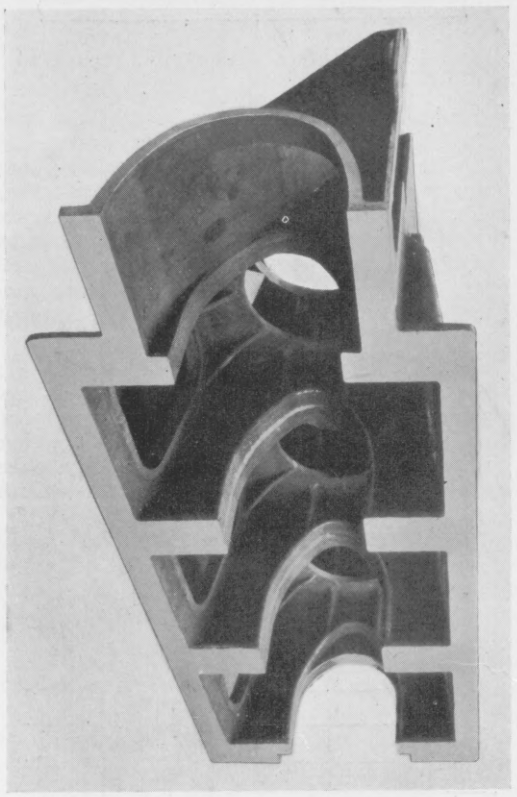

Fig. 48. Transverse webbing of box-frames

The nut itself is generally turned by means of a forged box-wrench, preferably one with a large-sized eye in the end of the handle. One hook of a 1½-ton chain falls is slipped into the eye, and the other is secured to some part of the engine-room casing by means of a beam-clamp. While a gang of men keeps a hard strain on the end of the wrench by means of the falls, a man with a strong back and a steady eye strikes the wrench with a 20-lb. sledge. Plenty of powdered graphite and lubricating oil discourages "cutting" of the threads.

Integral-Cast Types

One of the apparent reasons why tie-rods were used on the engine shown in Fig. 50 seems to be that the joint face between the cylinder beam and bedplate is of insufficient area to allow for the location of an adequate number of bolts. Note, also, that this design shows a bedplate of record depth and that anything like a separate box-frame is absent altogether. The bedplate, in fact, has been extended up so far that the space ordinarily taken up by the box-frame is now occupied by it. The main bearings are located at the bottom of deep notches in the bedplate, access being provided in the form of windows or doors broken through the side-walls of the latter.

Much the same tendency is observable in the 3000-s.h.p. submarine Diesel engine illustrated in Fig. 52, having the frame skeleton cross-sectioned in Fig. 51. Here the bedplate has again been made of considerable depth, the box-frame proper is omitted and the cylinder castings are bolted directly to what is really the bedplate surface. Note incidentally that the cylinder water jackets (Fig. 51) are essentially rectangular and that they are joined together by means of bolting flanges possessing great strength. The latter tie the entire framework together in such a way as to make a single box-like structure of the engine and impart to it a degree of rigidity perhaps not so easily attainable if the cylinders are more or less individualized.

The engine illustrated in Fig. 55 may possibly be regarded as the climax of the tendency toward integral engine structures. From one point of view, of course, it is a

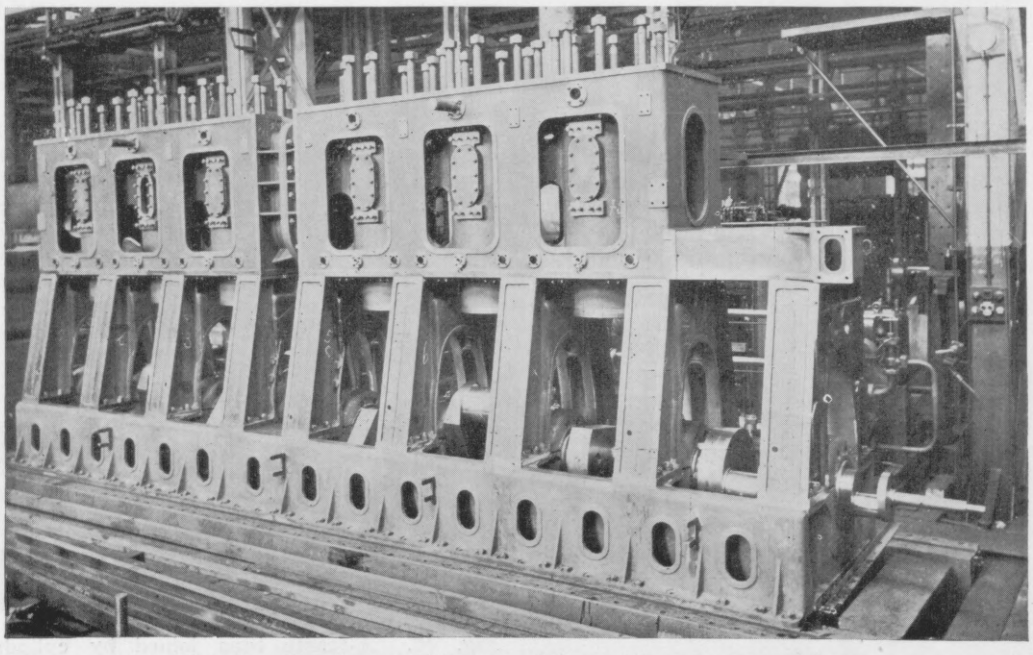

Fig. 49. Box-frame with separately-cast transverse members

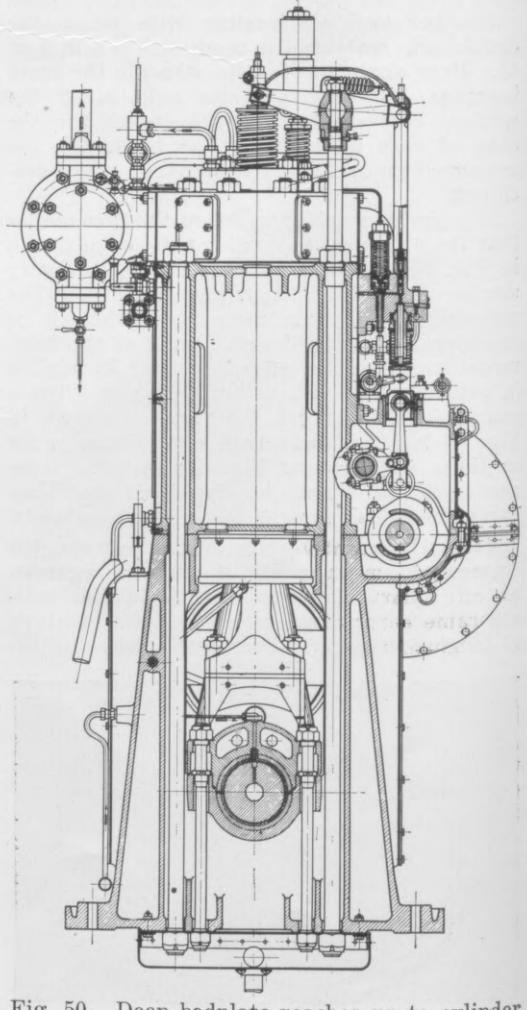

Fig. 50. Deep bedplate reaches up to cylinder level

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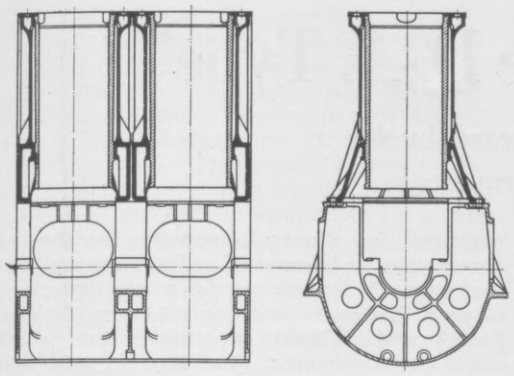
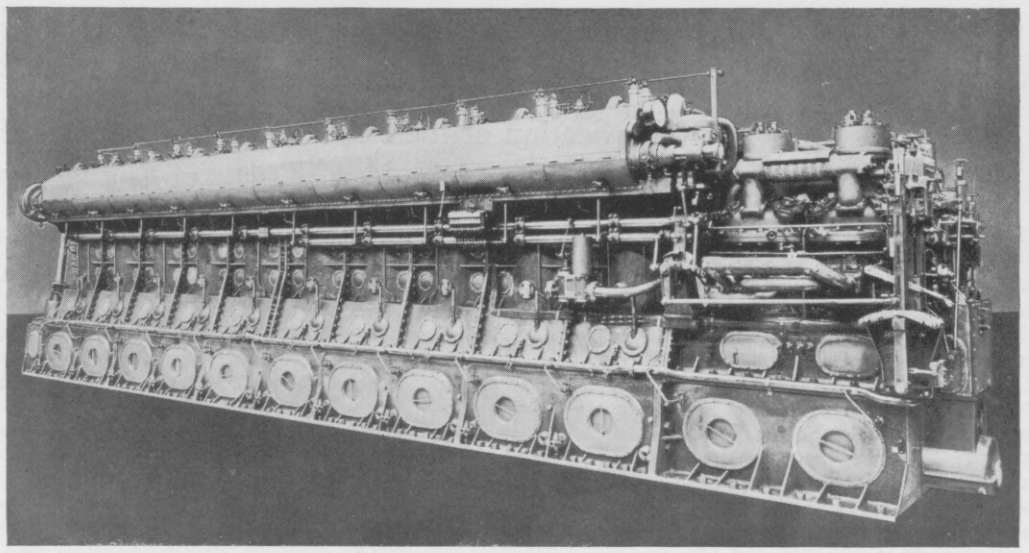


Fig. 51. Crankcase doors in sides of bedplate

repetition of the design previously considered, with the single difference that the bed-plate remains of normal depth (Fig. 54) while the cylinder castings are now extended downward so far that they embrace the crankcase. In a way that is true of the common A-frame shown in Fig. 40; but in the design now being considered the sidewise tying together is carried considerably further. Note that the brackets supporting the camshaft casing are also cast integral with the cylinder water jackets and that they provide the joint-faces for bolting the latter together. A further illustration of an engine design in which the cylinder mounting and crankcase constitute an integral casting resting on the bedplate is shown in Fig. 53.

What has been said about accessibiity in the preceding chapter applies with equal force to the types of framing which have just been gone over. In practically all of them forced lubrication has superseded drip feed and has required the closing off of the working parts for the sake of confining the oil spash. Bearing this fact in mind, designers have on the whole been guided by the idea of placing the metal of the framework in the locations where it would do the most good and have not asked



No. 52. Cylinder castings flanged together and to bedplate

very much as to whether it would prevent the operating engineer from seeing and touching the moving parts while they are actually in motion.

The engines shown in Figs. 52 and 55 illustrate this point particularly well, if indeed there should be need for dwelling on a matter

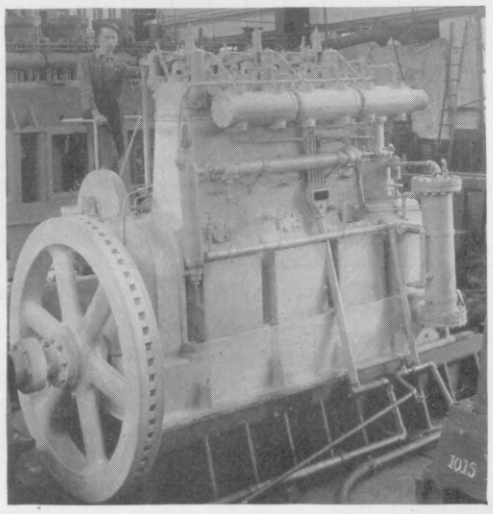


Fig. 53. Example of jointless framing

which was rendered more or less commonplace by experience with machinery other than oil engines. Who that drives an automobile of modern construction would think of opening the crankcase cover under the hood of his car and reaching in to feel the bearings? Probably from the first beginnings of automobile engine development that kind of messing around was ruled out by the very nature of things.

Both the manufacture and assembly of internal combustion engines must conform to higher standards than those which have in the past been found adequate for steam engines. In the case of the latter savings were probably achieved by the use of a flimsy open frame construction, the operating deficiencies of which were currently made good by the engineer on watch. Oil engines must be right before they are started up because splash lubrication and the nature of the frame constructions required generally rule out the possibility of setting anything right afterwards. Some might therefore say that the oil engine has sacrificed the convenience and accessibility of the steam engine; but there is also room for the opinion that modern oil engine designs have subjected designers to a new descipline and by means of it have caused them to make a noteworthy advance in the art of designing machinery.

One of the most valuable guides to trouble-shooting on Diesel engine work is contained in the section of the Motorship Yearbook (3rd edition) dealing with construction and operation of oil engines. The information is very condensed and therefore in very handy form. Engineers who are just beginning to study the Diesel engine will find a very instructive chapter on engine troubles.

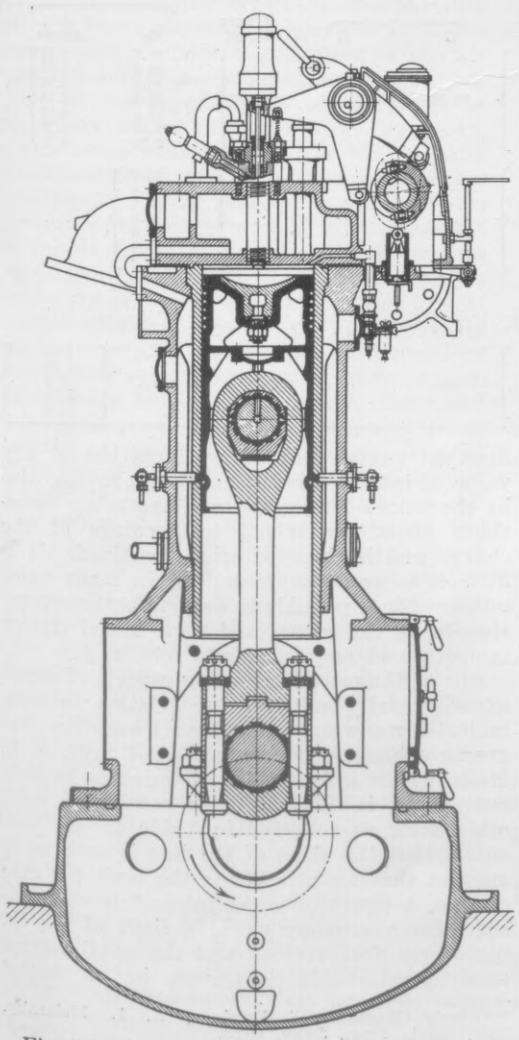


Fig. 54. Cross section of engine in Fig. 53

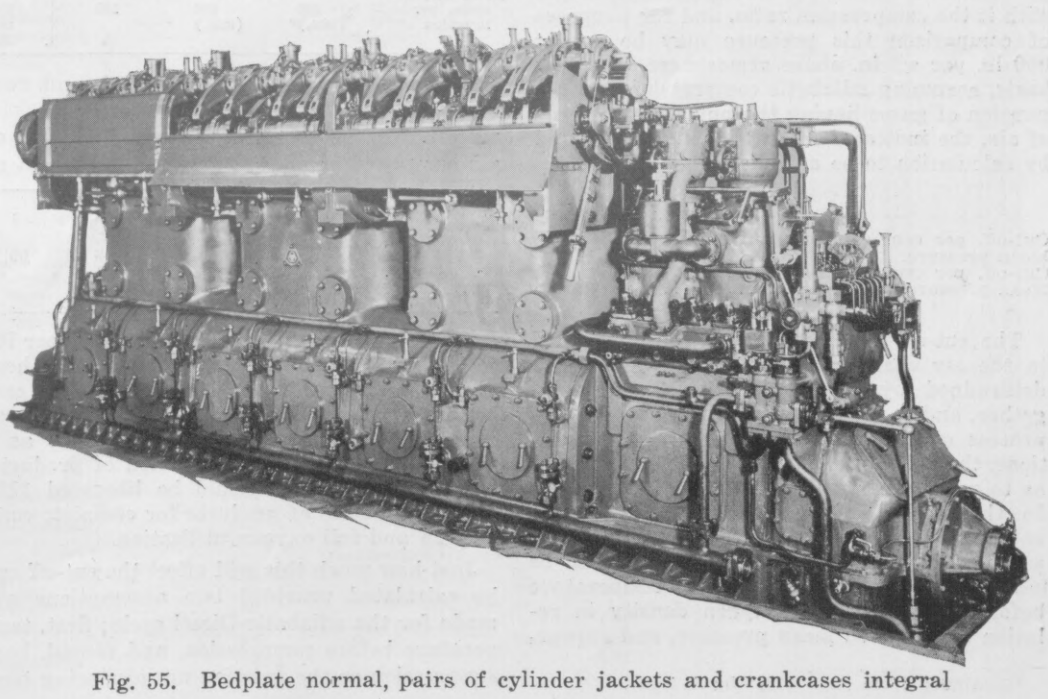


Fig. 55. Bedplate normal, pairs of cylinder jackets and crankcases integral

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Large Oil Engines, Especially the D.-A. Type*

This Section of the Paper Read by Dr. Chas. Edward Lucke Before the A. S. M. E. Treats of Engine Performances

WHILE the performance figures of various engines as published are often contradictory and confusing, there are some reports of tests of more than usual accuracy which may be made the basis of at least a tentative effort to clarify this situation. It is of considerable importance at this stage of progress in the development of large oil engines to establish such absolute quantities and such relations as may aid in selecting lines and which may be followed as most promising for development, if it should appear that any one were measurably better than another. The most important quantities to be examined in relation to each of the four possible styles of engine arrangement are indicated mean pressure with the mechanical losses in the engine leading to brake mean pressure, and indicated thermal efficiency with corresponding brake thermal efficiency and fuel consumption.

It has been pointed out that the really important thing in widening the scope of the economic utility of the oil engine is reduction of weight, and through it reduction of cost per horsepower on the one hand, and an increase in the horsepower per cylinder, preferably by the same means, if possible. Thermal efficiency is of secondary importance and may well be sacrificed to a reasonable degree, if by so doing something may be gained in the desired and necessary direction, but it would be especially satisfactory should it appear that the main ideals may be attained without loss of fuel consumption, and doubly so if some gain, however small, might also result. This really appears to be the case for the double-acting, two-cycle arrangement, and is one of the things that makes this direction of effort particularly interesting, even without utilization of the possibilities of raising mean pressures to high values, more or less equally available to all four of the typical arrangements—two- and four-cycle, single- and double-acting.

Indicated mean pressure of the thermodynamic cycle representative of the Diesel process of transforming heat of hot gases into work, is a function of the cut-off as in the uniflow steam engine, the diagram of which is identical except for the slope of the compression and expansion lines due to differences in fluids. It may be assumed that the maximum pressure is the same for all Diesel engines and with it the compression ratio, and for purposes of comparison this pressure may be set at 500 lb. per sq. in. above atmosphere. On this basis, assuming adiabatic compression and expansion of gases having the physical constants of air, the indicated mean pressures are found by calculation to be as follows:

DIESEL AIR-CYCLE MEAN PRESSURES

Cut-off, per cent displacement.....	3	6	9	12	15	18
Mean pressure, lb. per sq. in.....	17.15	34.16	50.76	67.50	83.95	99.70
Cut-off, per cent displacement.....	21	24	27	30	33	
Mean pressure, lb. per sq. in.....	116.0	132.4	148.36	164.19	179.64	

The cut-off is a function of the oil burned in the air charge in general, but actually is determined by several variables acting together, and is not determinable exactly by any process of calculation without many assumptions, the validity of which is so much in doubt as to make the results of very doubtful value. In the engine cylinder still other variables enter, including relations between variables assumed in thermodynamic calculations to be independent, such, for example, as temperature before compression or oxygen density in relation to indicated mean pressure, and appearance of carbon monoxide before oxygen in exhaust has been brought to zero, to cite only two.

It is of some interest, however, to set down limits to cut-off for assumed complete combustion of a fuel oil in air as one basis of comparison of actual cylinder performance, and to note its relation to incomplete utilization of air in terms of excess air or of free oxygen in the exhaust.

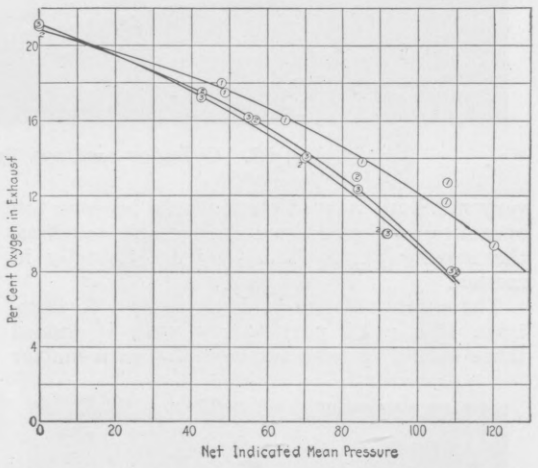

Fig. 17. Indicated mean pressure in relation to oxygen in exhaust

A fuel oil may be taken as more or less of the composition C = 0.85, H_2 = 0.12, and non-combustible = 0.03, ultimate analysis by weight. Its combustion requires 13.95 lb. of air per pound of oil, and its calculated heat of combustion, high value, is 18,660 B.t.u. per lb., when 14,544 and 60,626 are taken as the heat of formation of the products per pound of carbon and hydrogen, respectively, and the heat of formation of the fuel-oil compounds as 5 per cent of the heat of formation of products. On this basis there would be liberated 1250 B.t.u. per pound of products for complete combustion and full oxygen utilization.

Just how much this will affect the cut-off can be calculated provided two assumptions are made for the adiabatic Diesel cycle; first, temperature before compression, and second, heat suppression or the fraction not producing temperature rise during combustion because of dissociation, or because of radiation to walls at that time. It is sufficient for present purposes to note, however, that for cold air (60 deg. Fahr.) and no heat suppression the cut-off would be 33 per cent. For any value of heat suppression it would be smaller and in proportion to the effective heat, e.g., 22 per cent cut-off for one-third heat suppression. For higher initial temperature and for the same completeness of combustion, the cut-off would be shorter and inversely proportional to the absolute temperatures at beginning of compression. The oxygen in the exhaust products or the excess air resulting from non-attainment of as long a cut-off as might be possible will be zero for the above limiting cut-off of 33 per cent with 60 deg. Fahr. initial temperature and no heat suppression; and for higher temperature and some heat suppression, zero oxygen in products will correspond to shorter cut-offs. For any given initial temperature and heat suppression, oxygen will be inversely proportional to cut-off, and so also will be excess air.

There is as a result a definite relation between the excess or unused air or free oxygen in exhaust products, and the indicated mean pressure, both for the hypothetical thermodynamic cycle and for the working cylinder, similar qualitatively but different quantitatively. This fact makes a study of actual mean indicated pressures as related to free oxygen and excess air a matter of considerable importance, as showing not only how close to the limit an engine may be working, but at any value of indicated mean pressure showing also by the values for oxygen and excess air something about the initial temperature of the charge and heat suppression combined. It is difficult to separate these, but in many cases compared it is possible to assume heat suppression to be the same, and then initial charge temperatures may be judged relatively.

One of the most interesting reports of direct experimental results of the relation between indicated mean pressure (i.m.p.) and free oxygen in exhaust products is that of Prof. A. L. Mellanby[1] on a four-cycle air-injection engine, reproduced in Fig. 17 with results of some other tests as collected in Table 4. He concludes that the slope of the line is such as to suggest that i.m.p.= 140 is the limit for zero oxygen, a figure of great interest in comparison with a common working limit of 125 lb. per sq. in. for overload and the usual marine trial figure of 100 lb. per sq. in. A second

TABLE 4 INDICATED MEAN PRESSURE IN RELATION TO OXYGEN IN EXHAUST AND EXCESS AIR

Engine	Bore, in.	Stroke, in.	R.p.m.	I.m.p.	Oil C	Oil H	Air per lb. oil	Exhaust O_2	Exhaust Air, excess
Fraser and Chalmers, four-cycle, air-injection	21.5 in.	22.0 in.	300	108.0 85.4 64.8 48.3 49.0 107.7 120.5	82.96	11.87	13.75	12.7 13.8 16.0 18.0 17.5 11.65 9.39	1.42 1.82 3.06 5.79 4.92 1.18 0.77
Royal Tech. College, Mellanby and Gray	....	....	....	0 43 57 70 84 92 110	....	....	....	21.0 17.5 16.0 14.0 12.5 10.0 8.0	....
Four-cycle, air-injection	....	....	....	....	....	....	....	....	....
M. A. N. four-cycle, air injection	400 (mm.)*	600 (mm.)	160	117.0 97.7 77.5 58.8 38.5	....	....	....	....	1.57 1.89 2.5 3.63 5.0

*Continued from page 213, March issue.

[1] Clyde Marine Oil Engines, A. L. Mellanby, Inst. Mech. Eng., June, 1923.

(Continued on p. 295)

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(*Continued from page* 290)

curve, Fig. 18, completes the picture. The other test figures plotted are those reported by Prof. W. H. Watkinson[2] for a four-cycle, air-injection Fraser & Chalmers engine of new design, and by Prof. W. Riehm for a M. A. N. engine of the same class. These show differences in cylinder conditions clearly by the relation of one series of test values to another for equal values of i.m.p. Confirming the initial density effect on oxygen for a given i.m.p. value are the very unusual figures of Doctor Riehm for an engine tested with superchargers, 17.72 in. by 16.54 in. at 300 r.p.m. giving for values of i.m.p. 115.0, 146.5 and 176.0, with excess air of 1.86, 1.44 and 1.17, respectively

All of the previous figures are on four-cycle engines with air injection, which with sufficient spray air at a high enough pressure can agitate the air charge and drive the fuel charge through it sufficiently far to get good

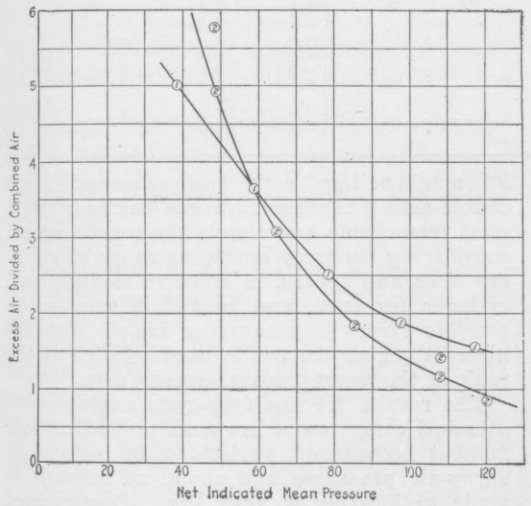

Fig. 18. Indicated mean pressure in relation to excess air

utilization in combustion chambers of the usual small range of shapes typical of single-acting engines. There might be a question as to the correspondingly effective utilization of the air in the rod end of a double-acting engine, but in view of the practicability of using two or more spray valves in such large cylinders, there is every reason to believe that equally good utilization of air can be secured, and this is confirmed by such reports as are available as to mean pressure attained. At the present time this is not so pertinent a matter as that of finding a construction that will not be injured by the high heat-generation rates typical of large cylinders with even the moderate mean pressures in common use corresponding to about half the air utilization, more or less.

Considering now the two-cycle cylinder and the possibility of developing indicated mean pressures as high as the four-cycle, it must first be noted that with the low air-utilization characteristic of four-cycle cylinders at normal rating values of indicated mean pressures, the primary problem in the two-cycle is not one of securing an equal air charge in view of low air-utilization limits imposed by the ordinary cast-iron metal walls, but merely one, as Doctor Naegel has pointed out, of conserving scavenging-pump work while supplying a sufficient air charge. This question, however, takes on a new interest in connection with the substitution of thin steel walls in place of cast-iron, as such walls can stand higher heat-generation rates per square foot, corresponding to higher indicated pressures. This fact makes the question of adequacy of air charge by two-cycle scavenging methods of greater importance for the future than for the past or present.

It has long been known by those investigating the two-cycle method of charging a cylinder with air that many of the untested ideas held by others concerning the problem were wrong. Nothing is easier than arranging inlet and exhaust openings and drawing arrows on paper to illustrate air flow and air displacement of products. The difficult thing is to make the air follow the arrows. Yet it is not so difficult to get a good air charge after all, and even with only a moderate expenditure of scavenging-air work, not much in excess of the so-called pumping loss of four-cycle engines equipped with valves of one-third of the cylinder diameter, more or less. Conversion of scavenging-air pressure to air velocity, forming entering streams of known velocity and direction, and the changing of direction of streams by wall faces, more or less equivalent to turbine vanes, so as to have the least short-circuiting of air from inlet to outlet, with maximum spreading of stream and displacement of burnt gases, constitutes the scavenging problem. This is not at all different in kind from that of room ventilation by incoming air streams or from the control of gases over heating and cooling surfaces, for best distribution. Lack of space forbids detailed discussion of this interesting subject here, but satisfactory conclusions are rapidly accumulating.

One such conclusion is to the effect that an excess volume of entering air is necessary and that the increase of scavenging ratio—pump to working-cylinder displacement—adds to the oxygen content remaining in the cylinder after exhaust ports close, up to a maximum of 1.6, more or less, with smaller and smaller gains as this limit is approached. High scavenging pressures are not necessary and, of course, are undesirable. In fact, the scavenging pressure may be so low as to make valve resistance a matter of quite as great importance as compression work, thereby leading to the abandonment of automatic valves in favor of mechanical ones and large ports. With large air volumes at low scavenging pressures it is entirely possible that the air-charge temperature in a two-cycle cylinder may be lower than in a four-cycle with its fixed clearance of 7 to 8 per cent full of burnt gases at exhaust temperature. This means greater density, and with oxygen content substantially or actually equal to that of pure air, a possibility of indicated mean pressures higher in two-cycle than in four-cycle engines over the effective stroke of the former or higher excess air and free oxygen for a given mean pressure less than the maximum, if for metal or other reasons the maximum is not to be used. Proof of the purity of the air charge in port-scavenged engines is being obtained by cylinder sampling apparatus, and the most complete reports on this subject are those of Dr. Naegel,[1] of Dresden. He finds for an Augsburg double-acting, two-cycle engine an oxygen content in the cylinder at the time of exhaust-port closure of 19 to 20 per cent at two points near the head and near the exhaust port, respectively, and for a Sulzer design of single-acting engine, 20.2 and 19.7 per cent, respectively. This is confirmed by Professor Nicholson's report of 19.3 per cent oxygen in the cylinder of a Scott-Still engine after scavenging.

This is substantially pure air, and with a proper scavenging ratio it may be cooler than is possible in a four-cycle cylinder.

It may therefore be concluded that for the effective stroke of the two-cycle cylinders, indicated mean pressures equal to those for four-cycle may be obtained, and with the same or greater unused air, if the value for i.m.p. used is less than the maximum.

Of the gross indicated horsepower, the part that becomes available as shaft or brake horsepower can be determined only by subtracting the losses in the engine. These losses are of three classes, representing the power absorbed, first, in the mechanism itself; second, in the compression of spray air; and third, in the pumping of the working air. For each of these there is an equivalent mean pressure, f.m.p., c.m.p. and s.m.p. or p.m.p., so that the net brake mean pressure is given for the various classes of engines by the following:

For two-cycle engines with attached air compressors and scavenging pumps,

$$b.m.p = i.m.p. - (f.m.p. + s.m.p. + c.m.p.) \quad [1]$$

For two-cycle engines with attached scavenging pumps but no compressors,

$$b.m.p. = i.m.p. - (f.m.p. + s.m.p) \quad [2]$$

For four-cycle engines with attached air compressors,

$$b.m.p. = (i.m.p.)\,G - (f.m.p. + p.m.p. + c.m.p.) \quad [3]$$

For four-cycle engines without attached compressors,

$$b.m.p = (i.m.p.)\,G - (f.m.p. + p.m.p) \quad [4]$$

The values for these various quantities are to be determined only from tests because no rational analysis can lead to reliable results, though such analysis is of value in examining experimental data. Ordinarily the indicated mean pressure as found experimentally is the net value, the four-cycle pumping loss being

[1] Loc cit.

[2] The Fraser & Chalmers Heavy Oil Engines, *Engineering*, June 6, 1924.

TABLE 5 I.M.P. AND B.M.P.—FOUR-CYCLE OIL ENGINES WITHOUT COMPRESSOR—GAS AND GASOLINE ENGINES

No.	Engine	R.p.m.	B.hp.	I.hp.	Cyl.	Bore, in.	Stroke, in.	B.m.p.	Net I.m.p.	F.m.p.
	Oil Engines									
1	Vickers, solid-injection	381.3	12.15	....	12	14.5	15.0	84.9	106.2	21.3
		119.8	....	....	6	24.5	39.0	80.1	102.0	21.9
2	Still, solid-injection	....	....	....	....	22	36	92.4	109.8	17.4
								84.3	98.5	14.2
3	Experimental, horizontal, single-cylinder	77.9	7.07	....	1	19	24	10.5	20.6	8.9
		101.0	19.1					21.85	34.6	9.3
		112.2	26.2					27.0	36.3	10.0
		142.1	53.8					43.7	53.9	10.8
		184.0	103.0					64.7	78.6	12.6
		216.0	124.7					66.6	78.6	12.0
		119.1	32.3					31.2	38.7	10.3
		141.4	53.6					43.7	48.0	11.4
		157.8	73.2					53.5	62.5	11.8
		180.2	100.9					64.7	77.8	12.7
		195.7	111.8					65.9	77.2	12.7
		223.5	129.1					66.6	80.5	11.8
		225.0	130.0					66.6	75.3	11.8
		247.0	142.7					66.6	77.7	11.2
4	British Admiralty exptl. submarine engine, air-injection without compressor	309.0	268.0	310	....	20.0	20.0	109.0	126.0	17.0
		282.0	212.0	254				95.0	114.0	19.0
		262.0	173.0	207				83.6	100.0	16.4
		245.0	138.0	174				71.5	90.0	18.5
		222.0	106.0	135				60.0	76.0	16.0
		....	314.0	373				126.0	150.0	24.0
		....	313.0	358				126.0	144.0	18.0
		....	288.0	344				127.0	150.0	23.0
		....	284.0	322				126.0	143.0	17.0
		....	283.0	334				126.0	149.0	23.0
	Gas Engines									
A	Crossley	170.0	....	....	....	21.0	30.0	76.4	88.7	12.3
		240.0	....	....	....	12.0	18.0	75.2	86.5	11.3
B	Tangye	190.0	....	....	....	16.5	23.0	68.7	79.5	10.8
C	Körting	162.0	....	....	....	16.92	27.55	61.6	72.5	10.9
D	Ehrhardt-Sehmer	150.0	....	....	....	24.4	29.52	57.3	69.0	11.7
E	Westinghouse, D A	149.0	....	....	4	23.5	33.0	79.0	99.0	20.0
F	Allis-Chalmers	83.0	....	....	4	42.5	54.0	38.6	53.0	14.4
	Gasoline Engines at peak of B.m.p.—R.p.m. Curve									
a	Liberty	1400	....	....	12	5.0	7.0	119.5	131.3	11.8
	Liberty	1400	....	....	12	5.0	7.0	132.0	141.8	9.8
	Liberty	1600	....	....	12	5.0	7.0	132.0	144.2	12.2
b	Packard	1200	....	....	8	4.75	5.25	132.0	145.8	13.8
		1400	....	....	12	4.75	5.25	124.5	139.1	14.6
c	Hispano-Suiza	1200	....	....	8	4.72	5.11	118.5	131.1	14.6
		1400	....	....	8	5.51	5.91	128.0	147.0	19.0

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automatically subtracted in planimetering the indicator card, and this, the only value available as the pumping loss, is normally not reported. On this basis the four-cycle relations are changed by introducing i.m.p. = (i.m.p.) G—p.m.p., so that

For four-cycle engines with attached air compressors,

$$\text{b.m.p.}=\text{i.m.p.}-(\text{f.m.p.}+\text{c.m.p}) \qquad [5]$$

For four-cycle engines without attached compressors,

$$\text{b.m.p.}=\text{i.m.p.}-(\text{f.m.p.}) \qquad [6]$$

Test data are available for some engines of these different classes which are sufficiently complete for an evaluation of all the above variables, though, of course, such tests are not yet so numerous as to fix these values with the conclusiveness that is desirable. Some of these test figures have been collected in Tables 5, 6, and 7 for three of the four classes of engines, and these are used as the basis of the determination of mechanical losses in each type of engine.

As scavenging pumps of two-cycle engines vary in ratio of displacement of pump to that of working pistons, in scavenging pressure, and in valve resistance, the first step in the analysis of two-cycle engines with attached scavenging pumps is to eliminate s.m.p as a variable. This can be done when the test data include separate determinations of scavenging indicated horsepower from which to determine s.m.p. and (b.m.p.+s.m.p.). The next step is similar in kind but concerned with air compression and determination of the (b.m.p.+ c.m.p) for four-cycle engines, or with the above (b.m.p.+s.m.p.+c.m.p.) for two-cycle engines.

When such steps have been taken and the results plotted, i.m.p. against (b.m.p.) or (b.m.p.+s.m.p.) or (b.m.p.+c.m.p.) or (b.m.p.+ s.m.p.+c.m.p.) as the case may be, the points always lie in a straight line.

This line has the general equation

$$\text{b.m.p.}=b\times\text{i.m.p.}-a \qquad [7]$$

This is equivalent to a friction mean pressure of

$$\text{f.m.p.}=(b-1)\times\text{i.m.p.}+a \qquad [8]$$

In this expression the constant a represents a part of the friction that is independent of the cylinder pressures, while the first term represents the rest of it, an amount varying with the cylinder pressure determined by the constant $(b-1)$, which is the slope of the line.

Considering the second term it can be shown that if A is its value for the four-cycle mechanism with compressor, then $\frac{A}{2}$ will be the equivalent for the two-cycle mechanism, neglecting an excess d of scavenger friction over that of a four-cycle valve gear. A corresponding addition e for the compressor mechanism will give the result for air-injection engines. Similarly, for double-acting engines the value $\frac{A}{2}$ applies to the four-cycle and $\frac{A}{4}$ to the two-cycle. Expressing by $B=(b-1)$ the slope of the line, which careful plotting and fairing of lines indicate is the same for all, the friction of the several types may be expressed by the following, introducing a term r for the friction of a second set of piston rings.

For single-acting, four-cycle engines without compressor,

$$\text{f.m.p.}=A+B\times\text{i.m.p.} \qquad [9]$$

For single-acting, four-cycle engines with compressor,

$$\text{f.m.p.}=A+e+B\times\text{i.m.p.} \qquad [10]$$

For single-acting, two-cycle engines with scavenging pump but no compressor,

$$\text{f.m.p.}=\frac{A}{2}+d+B\times\text{i.m.p.} \qquad [11]$$

For single-acting, two-cycle engines with scavenging pump and compressor,

$$\text{f.m.p.}=\frac{A}{2}+d+e+B\times\text{i.m.p.} \qquad [12]$$

For double-acting, four-cycle engines with compressor,

$$\text{f.m.p.}=r\left(\frac{A}{2}\right)+e+B\times\text{i.m.p.} \qquad [13]$$

For double-acting, two-cycle engines with scavenging pump and compressor,

$$\text{f.m.p.}=r\times\left(\frac{A}{4}\right)+d+e+B\times\text{i.m.p.} \qquad [14]$$

The experimental results reported in Table 5 for the four-cycle engines without compressors are plotted in Fig. 19. The fair line for vertical oil engines at constant speed is located, and from it the values of the constants are found to be $A=14.8$, $B=1.0$.

Here the British Admiralty tests by Hawkes are especially valuable, because, in addition to more than usual accuracy, the spray air was supplied by a separate compressor and the mean cylinder pressures run high but include a considerable range. Each test point is indicated and its relation to the fair straight line is clear. On this sheet are plotted points from the table for other engines—a horizontal oil engine at variable speed, horizontal gas engines, and vertical aircraft gasoline engines, all single-acting, with two cases of large double-acting tandem twin gas engines. It is most remarkable how closely the results agree, considering the truly enormous range of cylinder sizes and speeds, in addition to fuels and cylinder pressure, and judged in relation to possible errors in measuring i.m.p. For the aircraft engine the points were selected at the peak of the mean-pressure-speed curve.

The results for the four-cycle engines with attached compressors are next plotted in Fig. 20, but because of variations in values of spray-air pressures and c.m.p. no attempt is made at locating a fair line. Compressor-work data are not available for but two, the M.A.N. and the Fraser-Chalmers, and is different for these. To locate the friction line, the line for four-cycle engines without compressors from Fig. 19 is reproduced in Fig. 21, on which are located the points for these two tests, plotting vertically the sum b.m.p. and c.m.p, and locating the fair line representing the friction of the mechanism with attached compressor.

On the axes of i.m.p. the difference between the intercepts of these lines gives the value e, the constant mean pressure equivalent to friction of the compressor mechanism, the slopes

TABLE 6 I.M.P. AND B.M.P.—FOUR-CYCLE OIL ENGINES WITH ATTACHED COMPRESSORS

No.	Engine	R.p.m.	B.hp.	I.hp.	Cyl.	Bore, in.	Stroke, in.	B.m.p.	Net I.m.p.	C.m.p.	B.m.p. + C.m.p.
1	Beardmore	124	1267	....	6	24.4	38.4	75.5	98.0	....	....
	Tosi	129.6	1332	....	6	24.4	38.4	76.0	100.5	....	....
2	M. A. N.	190	....	....	....	11.3	18.1	88.2	106.0	....	....
								73.2	93.0		
								43.3	66.0		
3	M. A. N.	100	....	....	6	27.5	47.3	52.8	65.7	....	....
								53.6	67.0		
1	Beardmore-and Wain	123	....	....	8	23.2	31.6	70.5	90.3	....	....
								69.8	89.5		
5	Werkspoor	125	....	....	6	22.1	39.4	77.0	98.7	....	77.0
6	Mannheim	120	1600	....	6	23.6	39.4	77.0	90.5	....	....
7	M. A. N.	158	79.6	103.4	....	15.75	23.6	87.1	113.5	5.66	92.8
		159	67.4	89.4				73.5	97.5	5.46	79.0
		161.5	50.3	72.0				54.2	77.5	5.06	59.3
		163.0	33.0	55.0				35.2	58.8	4.92	40.1
		164.5	15.6	36.3				16.6	38.5	5.07	21.7
8	Mirrlees	250	....	....	....	12.0	18.25	80.7	109	....	....
		250						70.0	97		
		250						46.7	73		
		250						23.3	48.6		
		250						7.8	33.8		
9	Mirrlees	170	....	....	4	18.9	26.8	79.1	104.0	....	....
								75.6	101.0		
								59.3	82.0		
								58.7	82.0		
								40.8	63.4		
								79.8	110.0		
								77.2	108.0		
								58.8	86.0		
								58.1	86.0		
								41.1	68.0		
10	Fraser-Chalmers	300	1000	1306.6	....	21.5	22	82.6	108.0	8.1	90.7
		299.8	749.7	1033.1				62.0	85.4	7.8	69.8
		301.6	502.7	1788.1				41.3	64.8	7.5	48.8
		301.3	251.0	586.3				20.7	48.3	7.0	27.7
		302.5	....	299.0				....	49.0	....	....
		300.5	1001.7	1305.1				82.6	107.7	8.1	90.7
		301.1	1154.4	1463.6				95.0	120.5	8.2	103.2

TABLE 7 I.M.P. AND B.M.P.—TWO-CYCLE OIL ENGINES WITH ATTACHED SCAVENGING PUMP AND COMPRESSOR

No.	Engine	R.p.m.	B.hp.	I.hp.	Cyl.	Bore, in.	Stroke, in.	B.m.p.	I.hp.	S.m.p.	C.m.p.	B.m.p. + S.m.p. + C.m.p.	S.m.p. + C.m.p.
1	Sulzer	149.9	1791	....	....	39.4	43.5	89.4	104.0	....	....	....	....
	Sulzer	148.8	1986	....	....	39.4	43.5	99.75	111.5	....	....	....	....
2	Sulzer	101.6	1264.5	....	....	23.62	37.0	75.0	92.0	....	....	....	....
3	Sulzer	101.2	1262	....	....	23.62	37.0	75.4	103.5	....	....	....	....
4	Sulzer	290.0	550	....	4	340(m.)	540(m.)	85.0	124.0	....	....	....	....
			410	....	4								
5	Vickers	141.2	1042	....	....	30	36	114.7	124.1	....	....	....	....
6	Cammellaird-Fullagar	117.3	1037	....	4	18.5	25	65.0	78.1	....	....	....	....
	Cammellaird-Fullagar	117.3	....	1420	4	18.5	25	64.97	89.0	4.9	5.71	10.61	75.58
		118.3	....	1648				76.8	102.4	5.11	5.46	10.57	87.37
		58.5	....	473				36.29	59.5	1.23	6.45	7.68	43.97
		90.5	....	818				46.62	66.6	....	6.72	....	....
		110.6	....	1220				57.86	81.5	5.4	6.42	11.82	69.68
		121.1	....	1526				68.48	92.8	6.96	7.71	14.67	83.15
8	Cammellaird-Fullagar	120	....	....	4	18.5	25	62.0	90.8	....	....	....	....
		120						60.5	87.5				
		119						64.8	91.9				
		120						68.0	95.5				
		110						57.5	81.5				
		100						51.5	73.5				
		90						46.0	67.0				
		80						42.5	68.0				
		70						39.0	68.0				
		60						39.0	60.5				
								36.5	59.5				
9	Cammellaird-Fullagar	105	1000	1400	4	18.5	25.0	....	....	....	....	....	....
10	Bethlehem	116	2900	4000	6	25.5	48.0	67.5	93.0	....	....	....	....
11	Nobel	87.3	810	1137	....	26.58	36.22	45.1	63.1	2.91	5.05	7.96	53.06
		84.7	970	1290				55.5	74.9	2.90	5.16	8.06	63.56
		85.9	1232	1582				70.5	90.5	2.96	5.40	8.36	78.86
		87.4	1456	1797				81.1	100.1	3.17	5.50	8.67	79.77
		64.3	930	1142				70.2	86.2	1.91	5.05	6.96	77.16
		65.2	746	970				55.5	72.1	1.90	5.05	6.95	62.45
		65.9	578	770				42.7	57.0	2.05	4.76	6.81	49.51
		45.2	375	496				40.3	53.3	1.26	4.50	5.76	46.06
		106.6	948	1342				43.3	61.4	3.36	5.05	8.41	51.71
		106.0	1237	1623				56.6	74.3	3.36	5.30	8.66	65.26
		107.0	1635	2022				74.1	92.0	3.50	5.53	9.03	83.13
		105.1	1752	2128				81.1	98.5	3.48	5.65	9.13	90.41
		108.4	1958	2401				88.0	108.0	3.66	5.75	9.41	97.41
12	M. A. N. Exp'l D. A.	145	....	....	4	33.44	41.38	135.0	150.0	....	....	....	....

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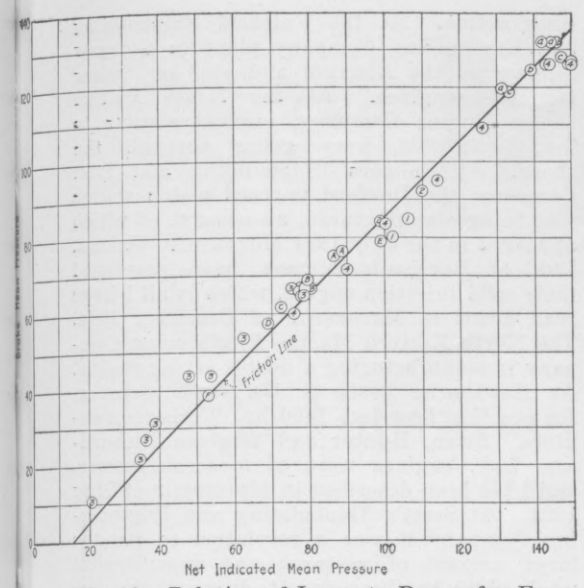

Fig. 19. Relation of I.m.p. to B.m.p. for Four-Cycle Engines without Compressors

of the two lines being unity. In this way the value is found $e = 1.9$.

This friction line for four-cycle engines with attached compressors is located on Fig. 20 for comparison.

Passing to the two-cycle engines there is a shortage of data for compressorless engines, but there are some good data for present purposes on engines with attached compressors and scavenging pumps. These results are plotted in Fig. 22. For these tests where both scavenging pump and compressor mean pressures are given, the values b.m.p.+c.m.p.+s.m.p. are plotted on Fig. 21 besides the value of b.m.p. for four-cycle engines without compressors, and the values of b.m.p.+c.m.p. for four-cycle engines with compressors. This checks the slope of the lines as unity for all styles of engines and determines the value of friction of scavenging-pump mechanism over four-cycle valve gears to be $d = 0.7$. This friction line is drawn in on Fig. 22 for comparison.

With these determinations of the constants the friction of single-acting engines may be regarded as at least tentatively established in a proper relation, one to another. It is furthermore possible, in the absence of test data, to approximate the value for double-acting engines with reasonable accuracy.

All of these constants have been evaluated except r. From all available information on four-cycle, single-acting trunk pistons it seems as if the pistons and rings accounted for some 70 per cent of the total constant mechanical friction, and that a set of rings represent about 15 per cent of the 70 per cent, or 10 per cent of the total in round numbers. Accepting this value $r = 1.1$ in the absence of any direct measurements on large engines, and adding up the various terms for the four classes of engines of major interest, the friction for each is as follows:

Four-cycle single-acting,

$$\text{f.m.p.} = A + e = 14.8 + 1.9 = 16.7$$

Two-cycle single-acting with attached scavenging pump,

$$\text{f.m.p.} = \frac{A}{2} + d + e = \frac{14.8}{2} + 0.7 + 1.9 = 10.0$$

Four-cycle double-acting,

$$\text{f.m.p.} = r \times \left(\frac{A}{2}\right) + e = 1.1 \times \left(\frac{14.8}{2}\right) + 1.9 = 10.04$$

Two-cycle double-acting,

$$\text{f.m.p.} = r \times \left(\frac{A}{4}\right) + d + e = 1.1 \times \left(\frac{14.8}{4}\right) + 0.7 + 1.9 = 6.67$$

For purposes of fair comparison of the four typical arrangements it is necessary to reduce values of scavenging and compressor mean pressures to a common fair value for all, which may be compared with such actual values as are available.

From the available test reports some figures of spray-air mean pressures are collected in Table 8, with indicated mean pressures in the working cylinders and the spray pressures used in atmospheres. Similarly, some corresponding values for scavenging mean pressures

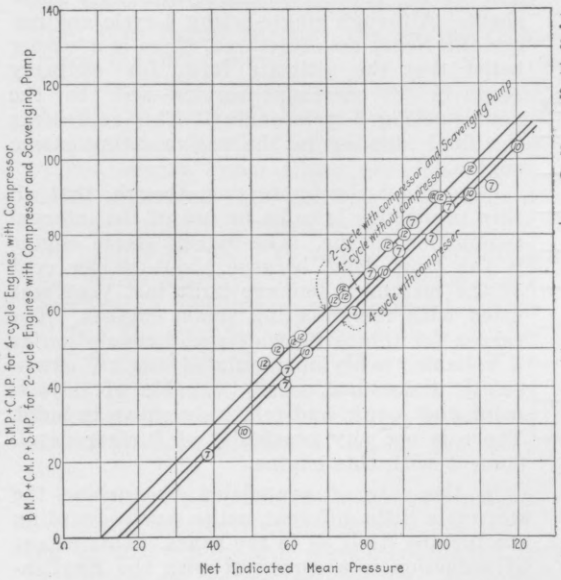
B.M.P. for 4-cycle Engines without Compressor
B.M.P.+C.M.P. for 4-cycle Engines with Compressor
B.M.P.+C.M.P.+S.M.P. for 2-cycle Engines with Compressor and Scavenging Pump
2-cycle with compressor and scavenging pump
4-cycle without compressor
4-cycle with compressor
Net Indicated Mean Pressure

Fig. 21. Friction of Single-Acting Oil Engines

are collected in Table 9, with scavenging pressures and the corresponding working mean pressures. Both sets of results, plotted in Fig. 23, show considerable irregularities, with, however, a tendency toward constancy or independence of both c.m.p. and s.m.p. with reference to i.m.p. at values dependent on whatever spray-air and scavenging supply pressures are used.

This is a natural result of absence of regulation of delivery pressure by means that reduce mean pressures appropriately, as may be done but not yet practised.

It is of course possible to calculate the mean pressures pretty closely for any assumed condition of valve resistance and delivery pressure for scavenging pumps, and get its equivalent s.m.p. referred to the working pistons through the cylinder ratios (displacement), and to do the same thing for three- or four-stage air compressors, but for purposes of type comparison of the four engine classes a flat assumption is made consistent with facts and indicated by the heavy lines on Fig. 23.

For scavenging air the value is taken as the high value reported by Prof. A. Rosborg[1] for the Nobel engine tested by him on the assumption of no regulation and s.m.p.=3.6 lb. per sq. in.

For spray-air compression the value taken is just above the points for the M. A. N. engine and assumed to be a straight line. It is represented by c.m.p.= 4.8 + 0.008 i.m.p., lb. per sq. in.

Collecting all of these losses, it is possible to set down values for the difference between indicated and brake mean pressures and for mechanical efficiencies E_m that, while not necessarily applicable exactly to any one engine, yet may be accepted tentatively as giving their

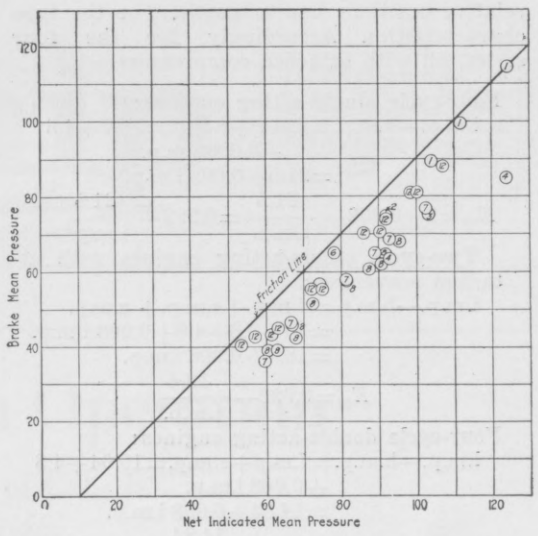

Fig. 22. Relation of I.m.p. to B.m.p. for Two-Cycle Engines with Attached Scavenging Pump and Compressor

TABLE 8—SPRAY-AIR MEAN PRESSURES

Engine	I.m.p.	Spray Pressure, atmos.	C.m.p.
Fraser and Chalmer, four-cycle	108.0	68	8.1
	85.0	61	7.85
	64.8	60	7.5
	48.3	60	7.0
	107.7	71	8.1
	120.5	75	8.2
M. A. N., four-cycle	113.5	63	5.66
	97.5	60	5.46
	57.5	50	5.06
	58.8	46	4.92
	38.5	46	5.07
Fullagar, two-cycle	89.0	..	5.71
	102.4	..	5.46
	59.2	..	6.45
	66.6	..	6.72
	81.5	..	6.42
	92.8	..	7.71
Nobel, two-cycle	61.1	48	5.05
	74.9	50	5.16
	90.5	55	5.40
	100.1	57	5.50
	86.2	48	5.05
	72.1	48	5.05
	57.0	42.5	4.76
	53.3	38	4.50
	61.4	48	5.05
	74.3	53	5.30
	92.0	59	5.53
	98.5	63	5.65
	108.0	67	5.75

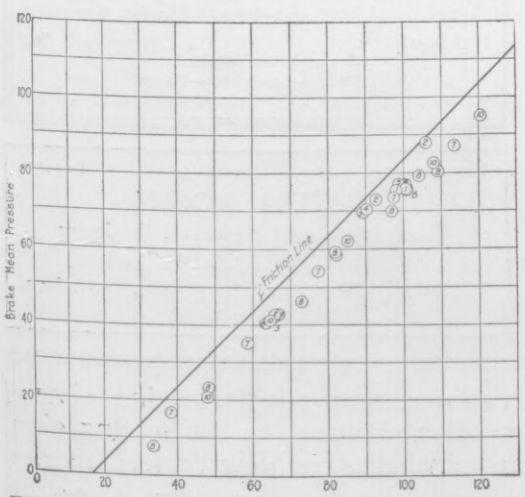

Fig. 20. Relation of I.m.p. to B.m.p. for Four-Cycle Engines with Attached Compressor

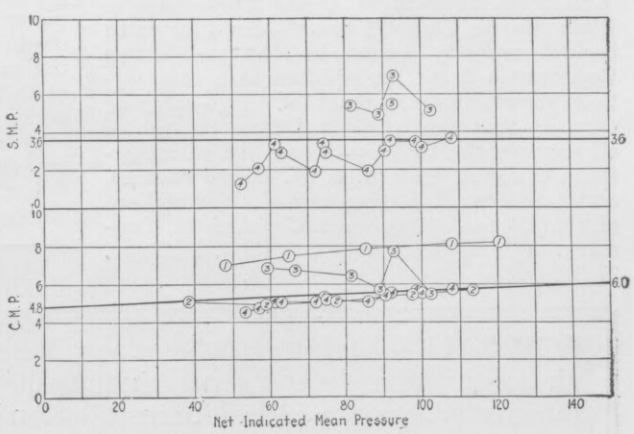

Fig. 23. Scavenging and Compressor Mean Pressures

[1] 1600 B.HP. Nobel-Diesel Marine Engine, George J. Steinheil, *The Engineer*, January 27 and February 3, 1922.

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relative positions one to another, or the type characteristics. Accordingly for the four styles, all with attached compressors—

Four-cycle single-acting engines:

$$\text{i.m.p.—b.m.p.}=\text{f.m.p}+\text{c.m.p.}=16.7+4.8+0.008\ \text{i.m.p.}$$
$$=21.5+0.008\ \text{i.m.p.}$$
$$E_m=1-0.008-\frac{21.5}{\text{i.m.p.}}=0.992-\frac{21.5}{\text{i.m.p.}}$$

Two-cycle single-acting engines with attached scavenger:

$$\text{i.m.p.—b.m.p.}=\text{f.m.p.}+\text{s.m.p.}+\text{c.m.p.}$$
$$=10+3.6+4.8+0.008\ \text{i.m.p.}$$
$$=18.4+0.008\ \text{i.m.p.}$$
$$E_m=0.992-\frac{18.4}{\text{i.m.p.}}$$

Four-cycle double-acting engines:

$$\text{i.m.p.—b.m.p.}=\text{f.m.p}+\text{c.m.p.}=10.04+4.8+0.008\ \text{i.m.p.}$$
$$=14.84+0.008\ \text{i.m.p.}$$
$$E_m=0.992-\frac{14.84}{\text{i.m.p.}}$$

Two-cycle double-acting engines:

$$\text{i.m.p.—b.m.p.}=\text{f.m.p.}+\text{s.m.p.}+\text{c.m.p.}$$
$$=6.67+3.6+4.8+0.008\ \text{i.m.p.}$$
$$=15.07+0.008\ \text{i.m.p.}$$
$$E_m=0.992-\frac{15.07}{\text{i.m.p.}}$$

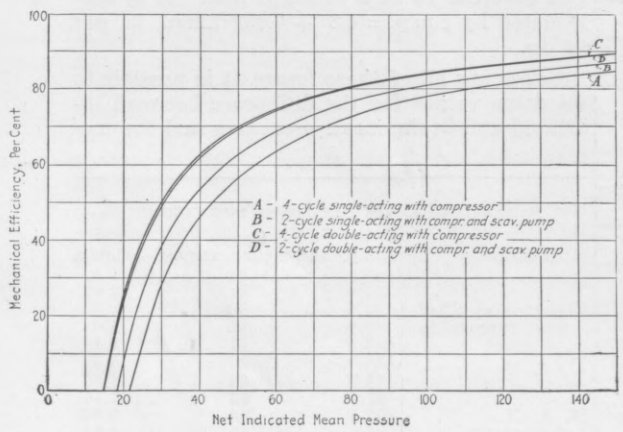
Mechanical Efficiency, Per Cent
Net Indicated Mean Pressure
A - 4-cycle single-acting with compressor
B - 2-cycle single-acting with compr and scav pump
C - 4-cycle double-acting with compressor
D - 2-cycle double-acting with compr and scav pump

Fig. 24. Mechanical efficiencies

These values are plotted in Fig. 24, and show clearly the relative positions of the four classes, the most interesting conclusion being the superiority in mechanical efficiency of double-acting over single-acting engines.

(*To be continued*)

Catalogs Received

Bulletin 35 describing the Falk Bibby flexible couplings and giving a list of the sizes and ratings. The Falk Corp., Milwaukee, Wis.

"A Few Advantages of the Burmeister & Wain Marine Diesel Engine"—An illustrated booklet recently issued by the well-known Copenhagen firm. In addition to a brief article showing the increase of cargo-carrying capacity that can be derived from the use of oil engines, the book contains a list of motorships equipped with B. & W. Diesel engines, and a number of illustrations of ships and engines. Burmeister & Wain, Copenhagen, Denmark. (U. S. Representative, H. C. Hallings, 27 Whitehall Street, New York).

British Oil Engine Industry

BY CHARLES F. GROSS, B.SC.*

To read about motorship construction and Diesel engine manufacture and then to see them are two different things. It was my privilege last summer to attend the meetings of the Institution of Naval Architects which met in London, followed by other meetings of the Institution of Engineers and Shipbuilders in Glasgow, and of the North-East Coast Institution of Engineers and Shipbuilders at Newcastle-on-Tyne, England. I visited twenty-one of the leading shipyards in Great Britain and all the leading plants where marine Diesel engines are being manufactured. I say manufactured, because when we use that word we usually think of quantity production.

Despite its increasing popularity, the marine oil-engine seems still to be in its stages of development there. Both as regards weight and cost of manufacture it leaves much to be desired. The question of weight of the engine will have to be seriously tackled. It must not be forgotten that weight roughly corresponds to manufacturing costs, and as things are at present, despite the unquestioned fuel economy of the heavy-oil engine, a reduction in the cost of manufacture must come about. Although single-acting 4-cycle engines are still being manufactured, there is a strong belief that the ultimate form for ordinary ships in the merchant service will be the double-acting 2-cycle engine. This represents the final step as in the reciprocating steam engine.

Do not get the impression, though, that all ship propulsion is to be by use of the internal combustion engine. The marine steam engine is far from dead, because about 50 per cent of the merchant tonnage built last year was fitted with reciprocating steam engines. The reason for this is its widespread reputation as a reliable, easily manipulated engine, extensively understood, easily capable of adjustment and repair and relatively cheap to build. There is not any possibility of further development with this engine.

In the case of propulsion by turbine the story is a little different, not so much regarding the turbine itself as in the gears. This means of reduction is a sore spot with the Englishman, because there has been an epidemic of failures with gears, especially with the double reduction type, although in their navy, where the single reduction gear only is employed, the record of the turbine-driven vessel has been satisfactory.

I bring in the question of steam propulsion at this point by way of comparison, to show that more progressive steps have been recently made in the development of the Diesel engine than in steam installations. The horsepower of the internal combustion engine has increased to amounts considered quite impracticable ten years ago. They are now building twin-screw vessels fitted with engines of 20,000 s.hp. And what is more remarkable, piston speeds of 1200 ft. per min. are being used. This speed might be contrasted with the rubbing speed of 620 ft. per min. attained at full load in the North British Diesel Engine which has reciprocating cylinders and pistons.

It is needless to give a detailed account of the engines which I saw under construction, but it may not be amiss to mention the yards where I spent some time looking over their construction. At the Fairfield Shipbuilding and Engineering Company plant in Glasgow I inspected the AORANGI and also her Sulzer two-cycle engines. At the North British Diesel Engine Company, engines similar to the SWANLEY'S were going through the shops. At Palmers Shipbuilding and Iron Company, the Doxford opposed piston engine was being manufactured, an account of which appeared in the July 1924 MOTORSHIP. Vickers, Ltd., at Barrow-in-Furness have developed their solid injection engine, which is all I have read about in MOTORSHIP of December 1924. The North Eastern Marine Engineering Company is manufacturing a double-acting engine. At Hawthorne Leslie & Co. there were on the erecting floor two 1400 hp. Werkspoor engines. Swan, Hunter and Wigham Richardson, Ltd., engines were of moderate power, and have been described in MOTORSHIP of May 1924. At Scott's Shipbuilding and Engineering Company it was a revelation to see the large amount of research and experimental work which has been done to turn out the machinery for the DOLIUS. Although such was needed it is of value in the repeat order which the Holt line has given to this company. Cammell-Laird & Co. are manufacturing the Cammell-Laird-Fullagar type, and I found this to be the only yard where they are attacking the problem of Diesel electric propulsion.

*Assistant Professor of Marine Engineering and Naval Architecture, Berkeley, Cal.

Under the leadership of the A.S.M.E. the week of April 20-25 has been selected for meetings in all the principal centers of the country to discuss problems related to gas engines and oil engines. All the leading national technical organizations have enrolled in the plan. The week is to be known as Oil and Gas Power Week.

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FOR SALE—McIntosh & Seymour Diesel Engine, 960 I.H.P. complete with all necessary auxiliaries, including air compressor, electric generating auxiliaries, pumps, oil tanks, air tanks, propeller, shafting, thrust bearing, oil purifier, spares, etc. Ideal for installation in a Tug Boat, Steamship, etc. Can be inspected in New York Harbor.
JOS. F. O'BOYLE - - - - - - - - - 15 Moore Street, New York City

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